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Code 8404 Naval Research Laboratory
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DIRECTOR NOTES

The readers of these notes may recall that I have previously written about Technology Transfer with respect to the role played by information analysis centers (IAC's). My thesis has been that the usefulness of IAC's as a mechanism for the transfer of technology from the public to the private sector has been greatly underestimated. I am encouraged that this situation is beginning to change.

This month I will participate in some of the workshops at the Spring Federal Laboratory Consortium on Technology Transfer hosted by the State of Oregon. I look forward to this as an opportunity to show how SVIC and other IAC's can form an effective bridge for information flow between the federal government and the state and local governments, as well as the private sector. The toll for crossing this bridge is surprisingly small when one considers the benefits of reaching the other side.

The implementation of federal policy on technology transfer has accelerated. The Federal Laboratory members of the Consortium do an excellent job of disseminating new technology through their monthly News Items and twice yearly workshops. As the central control point for technical information in a given discipline, an IAC can be an efficient link in the communication chain. We at SVIC expect to participate in this regard.

HCP

EDITORS RATTLE SPACE

PUBLIC SAFETY: WHAT RESPONSIBILITY FOR THE ENGINEER

The concern for public safety has grown in recent years. Many "unsafe" products are direct consequences of ever-expanding technology and consumer damand for more gadgets and conveniences. The engineer finds himself in the middle: product liability has become a real problem for engineers and designers. The number of lawyers who specialize in prosecuting accident cases involving huge damage settlements is increasing. Some of the accidents are attributable to misuse of the product, but others have occurred because of design defects. The level of safety of a product is usually not known. What can the engineer do to help himself and the consumer?

I believe that the engineer must include safety in his design criteria. In the past engineers were mostly concerned with function and structural integrity; reliability, life, and maintainability were less important considerations. In the end, function dictated design, after all, the device had to work. Now it's time to consider safety. Designing for safety -- whether to decrease noise or to make moving parts less accessible -- is not always difficult -- but it does increase production costs. And the consumer is going to have to accept that fact.

The engineer must also identify and assess the risks associated with a product he designs. The manufacturer, and ultimately the consumer, will have to evaluate these risks and the costs associated with safety. There will have to be a trade-off between cost and safety.

I am assuming that the design engineer is competent. Engineers can help themselves, the manufacturer, and the consumer if they avoid areas in which they lack familiarity with the latest technology or, plainly speaking, are incompetent. The worst situation is the engineer who fails to realize that he doesn't know enough to handle a specific mechanism or process. Such problems can be dealt with only through professional licensing and certification of competence. Too few engineers employed by manufacturers have a professional engineering license, and certification requires only college degrees. Because no continuing certification process exists, there is no way to guarantee that a competent engineer has designed a product. Many companies resort to extensive product testing; however, safety cannot always be evaluated with such testing because it is function oriented.

The product engineer can both protect himself and serve the consumer if he is competent in his field, is aware of safety criteria in the design process, and evaluates the risks associated with the use of any product.

R.L.E.

SPECTRUM

I have noted in the November 1976 **Shock and Vibration Digest** (Spectrum, pages 36 - 37), some comments by G. Harold Klein on my article "Vibration Isolating Mountings for Sensitive Equipment - New Design Criteria" (**Shock and Vibration Digest**, July 1976, pages 3 - 24).

I wish to thank Mr. Klein for his interest and constructive comment, and to reply to his adverse comment which I shall show is unjustified.

First, Mr. Klein quotes my statement, "It is remarkable that so many investigators of such a variety of systems continue to represent the equipment as a single lumped mass, thereby eliminating from the dynamic model the significant feature of the equipment -- its non rigidity." He assumes from this that I am unfamiliar with American literature on the subject and he recalls the now classic paper by R. D. Mindlin [1].

My article is a digest of two papers [2, 3] to which I referred the reader of the article for the background to the work described in the article. In Ref. [2] I named the well known American author the late C. E. Crede [4, pages 148, 299] as probably being the first to point out the significance of the natural frequency of a component of an equipment as a factor in deciding the natural frequency of a mounting for the equipment.

I thank Mr. Klein for reminding me of Mindlin's paper, which I had read and cited in other contexts some twenty years ago. Mindlin preceded Crede in discussing the response of a component of equipment on isolators subjected to excitation through the support and it was an oversight on my part that I omitted reference to Mindlin. But Mindlin cannot be credited with the results given in my article, which relate to a different problem as I show later in this communication.

In the two papers cited in my article eight of the nineteen references in [2] and three of the nine references in [3] are to American authors.

Clearly, then, I cannot concede that I am unfamiliar with the American literature on the subject. Furthermore my statement quoted above ". . . that so many investigators . . ." (I did not say " . . . all investigators . . .") is not in question for only three exceptions have been mentioned among the innumerable writers on vibration isolation over the past thirty years. Of these I cited one (Crede 1951) and Mr. Klein recalls another (Mindlin 1945). The third, mentioned by Klein, is Harris and Crede [5, Chapter 31, pages 20 - 27].

The second and more important criticism that I wish to refute is Mr. Klein's statement, "In summary then, Macinante's "Design Criteria" are certainly not "new", and should probably be credited to Mindlin." Earlier Mr. Klein states, in part, "Mindlin considered the identical problem as Macinante, namely the response of a component inside of a piece of equipment mounted on isolators." . . . . "The only difference is that Mindlin used a half-sinusoid forcing function rather than a damped sinusoid or sinusoidal forcing function."

Leaving aside the contradictory aspect of the last two statements, I shall now show that the problem solved and therefore the design criteria produced are different from those of Mindlin.

Mindlin's paper, which was motivated to develop the theory of drop testing of a packaged article, is in four parts, only one of which (Part III) is relevant to the present discussion. Part III includes the treatment of the response of a two-mass system representing the packaged article to a half-sinusoidal pulse of acceleration applied to the base of the container. Mindlin's design criteria relate the response of a particular component of the packaged article to a single frequency ratio (ω_2/ω_1) where ω_2 is the natural frequency of a component of the article and ω_1 is the natural frequency of the article on isolators (the Notation here is mine, not Mindlin's).

My criteria relate to the same two-mass system but with sinusoidal and damped sinusoidal forcing function and consequently I include the additional frequency ω_s (the excitation frequency). My results relate the response of a component of the equipment to two frequency ratios: (ω_1/ω_s) and (ω_2/ω_s). The difference between my results and Mindlin's is borne out by the fact that a surface representing Mindlin's results could have only a single 'resonance ridge' where $\omega_2/\omega_1 = 1$ (see Mindlin's Figs. 3.5.2 through 3.5.7 pages 432-436). A surface representing my results can have three ridges where $\omega_1/\omega_s = 1$, $\omega_2/\omega_s = 1$, $\omega_1/\omega_2 = 1$ as shown in Figure 8 of the article. In short, my criteria are 'new' in that they take account of the periodicity ω_s of the excitation.

Mr. Klein raises the important question of how the foundation (site) should be represented in the dynamic model. I agree with Mr. Klein that if one "measured the motion of the foundation prior to the installation of the equipment, the foundation motion will change after the equipment has been installed" and, in a later context, that ". . .if. . .the foundation is not rigid, possibly a three-degree-of-freedom model should be used, taking into account the change of the motion of the unloaded foundation due to the installation of the isolated equipment."

My reason for not increasing the complexity of the model in this way is that the analysis of even the two-degree system with damping and transient excitation leads to very cumbersome analytical expressions which after computation leave a formidable task of data reduction.

In my model the excitation is postulated to be a damped sinusoidal or a sinusoidal displacement of the site. In order to use my results for transient excitation the designer needs to make an estimate of the predominant frequency ω_s of the transient response of the site to impulses applied in the vicinity when the site is loaded by the installation. The literature contains adequate information on the dynamic behaviour of bases and foundations for the purpose of this estimation, which would be based on experimental data for the unloaded or partly loaded site. An accurate value of ω_s is not required, as the design data published in the article can quickly be used to determine the system response (transmissibility ratio) for the values ω_s covering the range of uncertainty. For steady state excitation, all that is needed is the predominant frequency of the applied excitation -- and this of course is independent of the site characteristics.

Finally, I fully agree with Mr. Klein's remarks on the difficulty of knowing what results and data exist and where to find them. Those of us working in the shock and vibration field have much less cause for complaint than those in many other fields, thanks to the excellent and growing services provided by the Shock and Vibration Information Center in matters of current awareness, information retrieval, reviewing and the Symposia.

REFERENCES

1. Mindlin, R.D., "Dynamics of Package Cushioning," Bell System Tech. Jour., 24, pp 353 - 461 (1945).
2. Macinante, J.A. and Walter, J.K., "The Isolation of Machine Tools from Site Vibration," Mech. Chem. Engr. Trans., I.E. Aust., MC9, pp 1 - 9 (1973).
3. Macinante, J.A. and Simmons, H., "Vibration Isolating Mountings for Sensitive Equipment - A More Realistic Design Basis," Mech. Chem. Engr. Trans., I.E. Aust., MC11, pp 22-32 (1975).
4. Crede, C.E., "Vibration and Shock Isolation," Wiley, New York (1951).
5. Harris, C.M. and Crede, C.E. eds., "Shock and Vibration Handbook," McGraw Hill, New York (1961, 1976).

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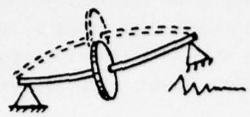
ROTOR-BEARING DYNAMICS: STATE-OF-THE-ART 1976



N.F. Rieger*

Abstract - This paper reviews recent developments in rotor dynamics and torsional dynamics of drive trains. The following topics are included: computer programs, balancing techniques, stability, and torsional dynamics of rotor systems. Attention is also drawn to a number of important remaining problems.

The various aspects of rotor-bearing dynamics problems have been understood for many years. Computer technology now allows realistic examination of these complex problems; in fact, the procedures have become design routines. Emphasis has thus shifted from creating new procedures for balancing and stabilizing rotors to refining existing ones.

COMPUTER PROGRAMS

Efficient computer programs are now widely available for calculating the following: static and dynamic properties of bearings, flexural critical speeds and mode shapes, synchronous unbalance response and transmitted force, stability of rotor systems, torsional critical speeds and transient response, and balancing of correction masses. Comprehensive reviews of such computer programs have recently been published [1, 2]. Most of these programs are available from Mechanical Technology Incorporated (MTI), Franklin Institute Research Laboratories (FIRL), and NASA - Computer Software and Information Center (COSMIC).

The most comprehensive source of bearing data [3] contains charts of dynamic stiffness and damping coefficients for many types of fluid-film bearings. Similar data are available for rolling element bearings [4]. A comparison of dynamic coefficient data for cylindrical bearings has been published [5], as have charts of dynamic coefficient data on specific bearing types [6 - 12]. Nonlinear bearing analyses are also available [13 - 16]. A variety of bearing dynamics programs are available; e.g., MTI Cadense series, FIRL programs, and certain NASA and COSMIC programs. A comprehensive review of bearing computer programs is needed.

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The many critical speed programs range from simple discrete mass/massless beam eigenvalue formulations to distributed mass-elasticity Timoshenko beam formulations. Techniques for calculating damped critical speeds have recently been developed. These programs have been discussed [1], and the theory has been given [17].

Rotor formulations vary from simple discrete mass/beam rotors to complex distributed mass-elasticity rotors. Bearing and support formulations include circular orbit two-coefficient bearings, eight- and 16-coefficient bearings [18], and the nonlinear bearing formulations mentioned above. The theory for modern mass unbalance response programs has been presented [19, 20]. Several simple programs are available from COSMIC; advanced programs are available from MTI and FIRL (Cadense and Rotdyn series). Complex programs for flexible rotors in flexible continuous supports have also been developed [21, 22]. Recent applications of the finite element method for unbalance response analysis have been developed [23, 24].

Stability programs -- available from MTI, FIRL, and Turbo Research, Incorporated (TRI) -- have been developed [16, 25, 26] and compared [1, 2]. Access to time-sharing programs is available through COM/CODE. Selection of a solution procedure depends upon the manner in which the stability threshold is sought and whether or not the bearing forces are linear or nonlinear. The rotor profile can be represented with all programs; the number of bearings varies from two for large nonlinear programs to 20 or 30 for efficient linear programs. The MTI and FIRL programs incorporate eight-coefficient linear representations of the bearings; the TRI program calculates the nonlinear bearing forces at each location in the orbit. The positive real part of the complex eigenvalue is sought for linear bearings; nonlinear bearing solutions plot the whirl orbit point-by-point and seek out dangerous growth conditions.

Several computer programs have been developed for balancing [27 - 29]. The influence coefficient theory

[30, 33] has been used in the exact-point-speed method and least squares method.

Torsional computer programs are available from MTI and FIRL that perform calculations ranging from natural frequency and mode analysis to transient dynamics of large damped nonlinear systems. Details about and comparisons of these programs have been published [1, 2].

BALANCING

Rigid rotor balancing has become a formalized routine: a balanced condition now depends more on the sophistication of the equipment used than on the skill of the operator. Automated high-volume balancing facilities have been developed for such rotor components as armatures and crankshafts. The International Standards Organization (ISO) has prepared a rigid rotor balancing standard document [31] that specifies maximum residual unbalance critieria for rigid rotors, operating speeds, and rotor weights. Flexible rotor balancing will be formalized when existing balancing procedures have been made more efficient. Standards for flexible rotor balancing are also being developed, and specifications for acceptable residual unbalance levels have been proposed [32].

Methods

Rigid rotors require balancing in one or two planes. Balancing instruments range from soft pedestal machines that indicate phase angle with a strobe-flash to hard bearing direct readout machines [33]. Field balancing of rigid rotor equipment has been described [33, 34], as have simple balancing techniques [35]. Most sophisticated industrial balancing equipment requires hard pedestal supports; other features of the equipment include automatic plane separation circuits, wattmeter signal filtration, and direct readout for magnitude and phase of required correction masses.

The most commonly used flexible rotor balancing techniques are the modal methods and the influence coefficient method. With modal methods residual unbalance effects are eliminated mode by mode within the operating speed range; the reintroduction of previously balanced modes is avoided. Several procedures are used; e.g., N modal method, N+2 method. Influence coefficient methods utilize computer procedures to process response data at selected speeds not necessarily near critical speeds. The required data set follows specified rules with regard to number of balance speeds, observations, and balance planes.

Modal balancing. The basic theory and developments of modal balancing have been analyzed [36 - 40]. Practical modal balancing techniques that utilize these principles have been described [41 - 43] for rotors used in large electrical equipment.

With another technique, called comprehensive modal balancing, the rotor is balanced in both rigid-body modes prior to flexible rotor balancing [44, 45]. Modal balancing and comprehensive modal balancing have been compared [46] for a uniform shaft in a high-speed balancing machine. Results indicate that the comprehensive modal method is superior. A computerized comparison [47] indicates that balancing the rigid-body modes may not be a general advantage and that rigid-mode correction mass effects might result in an inferior balance for speeds at which flexible rotor effects are important.

Influence coefficient balancing. Practical influence coefficient balancing of rotors began with Goodman [27]; Rieger [28] developed the procedure using a program by Lund. The method has been extensively tested [30, 48, 49] under exacting laboratory conditions. The significance of measurement errors and of correction weight installation errors has been studied [50, 51]. The influence coefficient method can be used through several flexural critical speeds. Two procedures have been developed: the exact-point-speed procedure and a least-squares procedure that accommodates additional input data. The effectiveness of the exact-point-speed procedure has been compared with the modal methods of Bishop and Gladwell [36] and of Moore [41] and with the comprehensive modal method of Kellenburger [43]. Two rotor systems -- medium steam turbine and small gas turbine -- were used to determine wide ranges of balance planes, speeds, and measurements. The influence coefficient method is generally superior to other methods. Similar conclusions have been reached by Dreschler [52], who sought an optimization of the influence coefficient method using a modal approach.

Balancing Machines and Balancing Standards

Rigid rotors can be automatically balanced to the extent necessary. All operations, including metal removal mass corrections, can be done automatically [33]. Computerized manual balancing allows corrections to be determined with push-button simplicity [53]. Field balancing has not yet been automated, but is has been computerized; terminals are now used to compute correction masses and to store data on similar problems for recall-guidance [54]. Laser balancing has been tested [54 - 56], but is not yet practical: initial costs are high, and the time required for metal removal in larger units is too long.

Flexible rotor balancing consoles now available contain a small computer and xy plotter to perform the balancing functions [54]. This procedure can be used for field balancing, either on site or on a dial-in basis to a computation center. Pedestal mounted-displacement sensors provide adequate signal information.

The ISO has recently issued several documents pertaining to the balancing of rigid and flexible rotors [31, 32]. The rigid rotor document [31] contains a glossary and a chart of acceptable residual unbalance versus speed for a wide range of rotor types -- from small ultracentrifuges to large propulsion gears. Flexible rotors have been classified into the five groups shown in Table 1.

Table 1. Classification of Flexible Rotors [32]

Rotor Class	Description	Example
Class I	Rigid rotors	Small rotor armature
Class II	Quasi-rigid rotors	Medium turbine rotor
Class III	Flexible rotors	Large generator armature
Class IV	Flexible rotors with flexible attachment	Turbine LP bladed rotor
Class V	Special rotors	Synchronous condenser

A discussion of various balancing procedures for each rotor class has been published [33].

STABILITY

Rotor stability studies have been concerned with improving the analytical capability for predicting linear instability threshold speed. Nonlinear programs have also been developed for predicting post-threshold behavior of the rotor. Linear analysis is now efficient and has been verified in the laboratory [56]. Nonlinear studies [13, 15] have improved the efficiency of the nonlinear approach and have been used to develop alternate formulations. An advanced nonlinear program [16] can accommodate a variety of system forces; e.g., nonlinear bearings, gas seal effects, nonsynchronous forcing, and residual unbalance. Table 2 shows the status of the various classes of rotor instability.

Basic experimental work is needed because many questions have not yet been answered [6]. For example, the basic instability mechanism is presently accepted to be described as an eigenvalue formulation. Smith [6] comments on three bearing instability mechanisms; Newkirk and Lewis [57] considered two mechanisms. Duckworth and Rieger [58] calculated threshold speeds from the results of several experimental investigations and achieved good threshold speed correlation. The linear stability program of Lund [54] has been verified under laboratory conditions. Apparently no similar tests have been undertaken, even though threshold speed correlation has been claimed in various industrial studies.

Linear Programs

Residual modal damping has been calculated for the rotor-bearing system at specified speeds [56]. The threshold speed is attained when the residual damping goes to zero in any mode. The complex eigenvalues are printed on a damped critical speed map that is associated with critical speed calculations for the rotor system. Complex eigenvalues of the characteristic matrix for the rotor-bearing system have been calculated [15, 18]. The instability threshold speed occurs when the real part of the eigenvalue changes from negative to positive. The imaginary part of the eigenvalue is the unstable whirl frequency of the rotor. This frequency corresponds to the lowest

Table 2. Classes of Rotor Instability

Nature of the Problem	Cures	Threshold Prediction Capability	Reason
Fluid-film bearing whirl	Increase eccentricity Tilting-pad bearings	Good	Reasonably good bearing data
Seal fluid whirl	Taper seals	Poor	Inadequate seal coefficient data
Structural hysteretic whirl	Nonrotating viscous damping	Poor	Inadequate technology of sliding fits
Rotor dissimilar stiffness	Tashing the rotor or otherwise improving symmetry	Medium	General studies; no specific programs
Rolling element bearing effects	Increased viscous damping	Poor	Mechanics not well understood
	Self-aligning bearings		No system studies

damped critical frequency of the rotor-bearing system. Efficient programs for both procedures are available.

Nonlinear Analyses and Techniques

Field experience suggests that many rotors operate in the post-threshold region with no adverse effects. Studies of such rotors are infrequently reported because the threshold region is associated with large whirl amplitudes [57, 59]. Real fluid-film bearings are highly nonlinear components. Representations of the nonlinearities have been examined [13 - 16, 60] and used to develop computer programs [6, 15, 16]. Acceptance of these programs has been limited because orbit calculations are expensive, and linear stability programs have been effective. Most nonlinear developments have involved digital computer programs. An analog study [15] examined the effects of a complex nonlinear short 180° plain bearing on the performance of a two-mass rotor. The complete model accounted for unbalance, internal hysteresis, and seal forces; parametric studies of system performance were also reported.

Tondl [63] first applied nonlinear vibration theory to the rotor-bearing problem in 1965. Early studies dealt with resonance vibrations of a rotor; nonlinear factors were accounted for. He identified conditions for subharmonic and ultraharmonic resonances. Certain solutions to the equations are obtained using an analog computer to solve them. Experimental studies supported these analog results.

Tondl has reported a general study of self-excited vibrations of one- and two-mass systems [64]. He reviewed such effects as dry friction and initial conditions on various states of stability. He studied resonance vibrations on nonlinear systems excited by a periodic nonharmonic force [65]. Tondl reviews methods of solution and then applies them to selected examples. Problems of self-excitation of rotors [66] are the subject of the third volume in this series; a variety of destabilizing effects -- external damping, nonlinear restoring force, and rotor unbalance -- on the equations of motion and on certain two-degree of freedom (rigid rotor) systems are discussed. Families of solutions are compared with experimental data from simple high-speed rotors in hybrid bearings. Interested readers should also consult other work by Tondl.

TORSIONAL DYNAMICS

Early torsional vibration analyses [67, 68] contained practical procedures based on tabular natural frequency calculations. A number of torsional dynamics computer programs have now been developed that have multiple gear mesh capability. Solution procedures range from extended Holzer techniques to time-step integration routines. These programs have made possible the solution of torsional dynamics problems involving forced vibration; shock, gear and coupling backlash; and other transient problems. Advanced problems to which such programs have recently been applied are shown in Table 3.

Table 3. Torsional Vibration Problems

Problem Area	Program Description	Calculated Results
Helicopter gearbox	Branched gear mesh program including damping; extended Holzer method	Forced response using gear profile Fourier coefficients
Rolling mill drive dynamics	Branched gear mesh with backlash	Time-displacement of inertia members of system
Turbine generator electrical transient loadings; faults, negative sequence currents	Bladed-disk torsional system with step transient input	Natural frequencies; mode shapes

Inaccuracies in gear profiles have caused torsional dynamics problems [69, 70]. Computer programs have been developed to predict the magnitude of the forced vibrations that can arise from such inaccuracies [71]. The tooth-to-tooth spacing variation is represented in terms of Fourier coefficients; the forced-damped vibration problem is solved by Holzer iteration. Modified programs include formulations for double planetary gearboxes; an acoustic radiation analysis has also been incorporated [72]. The anticipated noise level spectrum for a given helicopter or marine gearbox design can thus be predicted and compared with alternative designs. Test verification in helicopters has shown excellent correlation for system frequencies and reasonable correlation (3 to 10 db) for noise levels.

Backlash in geared systems and couplings can cause transient torques three to four times greater than the steady drive torque values [73]. Numerical routines have been developed to integrate the equations in time steps, so that stress cycles following transient loading can be traced. Details of this type of analysis and its application to several rolling mill systems have been described [74, 75].

An important problem concerns the torsional oscillations of turbine-generator sets excited by either sequential electrical faulting, negative sequence currents, or both. Generator-induced oscillations occur at twice the line frequency and may persist at sufficiently high levels to cause blade failures in low pressure turbine stages. Electrical faulting and the effect of torsional excitation on generators has recently been discussed [76] in a study of a potentially dangerous condition. Free vibrations of bladed disks have been studied [77 - 79]. Significant coupling generally occurs through the disk; blade modes occur as banded groups of frequencies [80, 81]. Prediction of system natural frequencies for a turbine generator is a complex problem that has not yet been studied in detail, even though experimental data have been accumulating for several years.

CONCLUSIONS

- The capability to predict the dynamic performance of a rotor-bearing system depends mainly on accurate information about the bearing properties and on details of the unbalance distribution. Existing analytical capabilities appear to be adequate for design, but additional bearing data are needed.
- Damped critical speed analysis is a valuable and efficient procedure for obtaining concise information about critical speeds of rotor systems and about sub-synchronous whirl stability of the system.

- Predictions of the threshold of instability of rotor-bearing systems is limited by lack of information about the dynamic properties of bearings. Linear results give no indication of post-threshold whirl behavior. Nonlinear programs can provide details about orbital and other sub-synchronous whirl effects.
- Rigid rotor balancing is a well established technology; fully automated high-volume rigid rotor balancing systems are now in use.
- Modal methods and the influence coefficient methods are effective for balancing flexible rotors. Industrial use of the influence coefficient method for shop and field balancing has expanded rapidly since the development of computerized balancing techniques.
- A variety of sophisticated and effective torsional dynamics programs can predict transient time histories and amplitude response.

PROBLEM AREAS

- The lack of sufficient data on dynamic coefficients of bearings continues to limit the accuracy of rotor-bearing system calculations. An experimental program to develop data for major bearing types over wide parametric ranges is needed, particularly data for large bearings in turbulent flow. Such data should be correlated with bearing program data to define further program needs.
- General balancing procedures should be developed for various classes of industrial rotors in order to establish a specific technology for each type. Such data could be stored and used to save time in future balancing problems.
- Techniques are needed to safeguard turbine-generator sets from electrical faulting, and from sustained dangerous excitations; e.g., negative sequence currents. Test data from known faulting conditions are needed. Analytical techniques for predicting blade vibration levels anywhere in the machine should be developed and verified.
- Stability problems of rotor systems having zero viscous damping and small hysteretic internal damping are not well understood. Simple stabilization techniques and a program for predicting instability threshold speeds based on an improved hysteretic model should be developed. Hysteretic calibration procedures for this model are also needed.
- The destabilizing effect of electromagnetic fields and current density should be examined further.

REFERENCES

1. Rieger, N.F., "Rotor-Bearing Systems," Structural Mechanics Computer Programs, Univ. Virginia Press, Charlottesville, pp 473-498 (1974). See also Rieger, N.F., "Unbalance Response and Balancing of Flexible Rotors in Bearings," ASME Flexible Rotor Systems Committee Monograph No. 3 (June 1973).

2. Shapiro, W. and Pilkey, W., "Computer Programs for Rotor-Bearing System Analysis," Chapter 11 in Shock and Vibration Information Center Monograph No. 10, Naval Research Laboratory, Wash., D.C. (Sept 1975).

3. Lund, J.W., "Rotor-Bearing Dynamics Design Technology, Part III: Design Handbook for Fluid Film Type Bearings," Mechanical Technology Inc. Tech. Rept. AFAPL-TR-65-45 (May 1965).

4. Lewis, P. and Malinowski, S., "Rotor-Bearing Design Technology Part IV: Ball Bearing Design Data," Mechanical Technology Inc. Tech. Rept. AFAPL-TR-65-45 (May 1965).

5. Thomas, C.B. and Rieger, N.F., "Comparison of Dynamic Bearing Coefficients for a Plain Cylindrical Fluid-Film Bearing," Tech. Memorandum 74 WRL M3 Rochester Institute Tech., NY (May 1974).

6. Smith, D.M., Journal Bearings in Turbomachinery, Chapman and Hall, London (1969).

7. Warner, P.C., "Static and Dynamic Properties of Partial Journal Bearings," J. Basic Engr., Trans. ASME, Paper No. 62 LUBS-11 (June 1963).

8. Lund, J.W. and Sternlicht, B., "Bearing Attenuation," Tech. Rept. Bureau of Ships, NObs - 78930, Gen. Engrg. Lab, G.E. Co., Schenectady, NY (1961).

9. Lund, J.W., "Spring and Damping Coefficients for the Tilting-Pad Journal Bearing," Amer. Soc. Lubric. Engr. Trans., 7, pp 342-352 (1964).

10. Gleinecke, J., "Experimental Investigation of the Stiffness and Damping Coefficients of Turbine Bearings and Their Application to Instability Prediction," *Instn. Mech. Engr. Proc., Symp.* Lubric. Wear, 20-22-76, 181, Part 3B (1966-67).

11. Morton, P.W., "Measurement of the Dynamic Characteristics of a Large Sleeve Bearing, " J. Lubric. Tech., Trans. ASME, Series F, 93 (1), pp 143-150 (1971).

12. Morton, P.W., "The Derivation of Bearing Characteristics by Means of Transient Excitation Applied Directly to a Rotating Shaft," IUTAM Symp. Dyn. Rotors, Lyngby, Denmark (Aug 12-16, 1974).

13. Holmes, R., "Oil Whirl Characteristics of a Rigid Rotor in Journal Bearings," Instn. Mech. Engr. Proc., 291 (11) (1963).

14. Mitchell, J.R., Holmes, R., and Byrne, J., "Oil Whirl of a Rigid Rotor in 360° Journal Bearings," Instn. Mech. Engr. Proc., 180, (25), Part 1 (1965-66).

15. Rieger, N.F. and Thomas, C.B., Jr., "Some Recent Computer Studies on the Stability of Rotors in Fluid-Film Bearings," IUTAM Symp. Dyn. Rotors, Lyngby, Denmark, pp 436-471 (Aug 12-16, 1974).

16. Giberson, M.F., Proprietary Computer Program, Turbo Research Inc., West Chester, NJ.

17. Lund, J.W., "Stability and Damped Critical Speeds of a Flexible Rotor in Fluid Film Bearings," J. Engr. Indus., Trans. ASME, Series B, 96 (2), pp 509-517 (May 1974).

18. Lund, J.W., "Computer Programs for Unbalance Response and Stability, Part V," AFAPL Tech. Rept TR-65-45, Wright-Patterson AFB, Dayton, OH (May 1965).

19. Lund, J.W. and Orcutt, F., "Calculations and Experiments on the Unbalance Response of a Flexible Rotor," First Vib. Conf., Boston, ASME Paper 67-VIBR-27 (1967).

20. Morton, P.W., "On the Dynamics of Large Turbo-Generator Rotors," Instn. Mech. Engr., Proc., 180 (12), p 295 (1965-66).

21. Richardson, J.W., Proprietary Computer Program, Mechanical Technology, Inc., Latham, NY (1968).

22. Rieger, N.F., Proprietary Computer Program, Rochester Institute Tech., NY (1972).

23. Ruhl, H. and Booker, J., "A Finite Element Model for Distributed Parameter Turborotor Systems," J. Engr. Indus., Trans. ASME, 126 (Feb 1972).

24. Zorzi, E.S. and Nelson, H., "Finite Element Simulation of Rotor-Bearing Systems with Internal Damping," ASME Gas Turb. Conf., New Orleans, Mar 1976 (To be published).

25. Lund, J., Proprietary Computer Program, Mechanical Technology, Inc., CADENSE 25, Latham, NY (1973).

26. Gunter, E.J., Franklin Inst. Res. Lab. Proprietary Computer Program ROTDYN, Philadelphia, PA (1972). See also: NASA Report No. NAS53-6473 "Dynamic Stability of Rotor-Bearing Systems" (1966).

27. Goodman, T.M., "A Least-Squares Method for Computing Balancing Corrections," J. Engr. Indus., Trans. ASME, Series B, 86 (3), pp 273-279 (Aug 1964).

28. Rieger, N.F., "Computer Program for Balancing of Flexible Rotors," Mechanical Technology Inc. Tech. Rept. 76-TR-68 (Sept 1967).

29. Gunter, E.J., Proprietary Computer Program, Dept. Mech. Engr., Univ. Virginia, Charlottesville (1973).

30. Tessarzik, J., Badgley, R.H., and Anderson, W.J., "Flexible Rotor Balancing by the Exact Point-Speed Influence Coefficient Method," Paper submitted to Third Vibrations Conf., ASME, Toronto, Paper No. 71-Vibr-91 (Sept 1971).

31. International Standard Organization Document, "Balance Quality of Rotating Rigid Bodies," ISO 1940, 1973.

32. International Standard Draft Document, "Mechanical Balancing of Flexible Rotors," ISO/TC 108/SC 1/WG 2 (SECR-1) 4.

33. Rieger, N.F., "The Balancing of Rigid and Flexible Rotors," Shock Vib. Info. Ctr. Monograph No. 12 (in preparation).

34. Langlois, A.B. and Rosecky, E.J., "Field Balancing," *Allis-Chalmers Mfg. Co., Milwaukee, WI* (Jan 1968).

35. Wilcox, J.B., Dynamic Balancing of Rotating Machinery, Sir Isaac Pitman Sons, London (1967).

36. Bishop, R.E.D. and Gladwell, G.M.L., "The Vibration and Balancing of an Unbalanced Flexible Rotor," J. Mech. Engr. Sci., 1 (1), pp 66-77 (1959).

37. Bishop, R.E.D. and Parkinson, A.G., "On the Isolation of Modes in the Balancing of Flexible Shafts," Instn. Mech. Engr. Proc., 177 (16), pp 407-423 (1963).

38. Bishop, R.E.D. and Parkinson, A.G., "Residual Vibration in Modal Balancing," J. Mech. Engr. Sci., 7 (1), pp 33-39 (1965).

39. Parkinson, A.G., "Balancing of Flexible Shafts Rotating in Massive Flexible Bearings," J. Mech. Engr. Sci., 15, p 430 (1973).

40. Parkinson, A.G., "The Balancing of Flexible Rotors," IUTAM Symp. Dyn. Rotors, Lyngby, Denmark, pp 413-435 (Aug 12-16, 1974).

41. Moore, L.S. and Dodd, E.G., "Mass Balancing of Large Flexible Rotors," G.E.C. J., 31 (2), pp 74-83 (1964).

42. Moore, L.S., "Balancing of Large Turbine Rotors," Inst. Marine Engr. Trans., 81, pp 15-115 (Apr 1969).

43. Moore, L.S., "The Significance of Anistropy of Support Conditions When Balancing Very Large Flexible Rotors," Vib. Rotating Sys., Conf. Instn. Mech. Engr., pp 86-95 (Feb 1972).

44. Kellenberger, W., "Should a Flexible Rotor Be Balanced in N or (N+2) Planes?," ASME Paper No. 71-VIBR-55 (1971).

45. Kellenberger, W., "Limits to Modal Balancing of Flexible Rotors," IUTAM Symp. Dyn. Rotors, Lyngby, Denmark, pp 206-243 (Aug 12-16, 1974).

46. Giers, A., "Comparison of the Balancing of Flexible Rotors Following the Methods Federn-Kellenberger and Moore," VDI-Berichte Nr, 161, pp 29-34 (1971).

47. Kendig, J.P., "Comparative Study of Methods for Flexible Rotor Balancing," MS Thesis, Rochester Institute Tech, NY (Nov 1975).

48. Badgley, R.H. and Tessarzik, J.M., "Experimental Evaluation of the Exact Point-Speed and Least-Squares Procedures for Flexible Rotor Balancing by the Influence Coefficient Method," ASME Paper No. 73-DET-115.

49. Badgley, R.H., Tessarzik, J.M., and Fleming, D., "Experimental Evaluation of Multiplane-Multispeed Rotor Balancing through Multiple Critical Speeds," ASME Paper No. 75-DET-73, Design Engr. Tech. Conf., Wash., D.C. (1975).

50. Lund, J.W. and Tonnesen, J., "Analysis and Experiments on Multi-Plane Balancing of a Flexible Rotor," ASME Paper No. 71-Vibr-74.

51. Tonnesen, J., "Further Experiments on Balancing of a High-Speed Flexible Rotor," ASME Paper No. 73-DET-99 (1973).

52. Drechsler, J., "A Combination of Modal Balancing and the Influence Coefficient Method," Ph.D. Thesis Abstract, Tech. Univ., Berlin (Apr 1975).

53. Schenck-Trebel Corp., "Universal Hard Bearing Balancing Machines," Brochure A-1102, Deer Park, Long Island (1972).

54. Badgley, R.H., "Recent Developments in Multi-plane-Multispeed Balancing of Flexible Rotors in the United States," IUTAM Symp. Dyn. Rotors, Lyngby, Denmark (Aug 12-16, 1974).

55. Schenck-Trebel Corp., "Laser Balancing Device," Balancing News (Spring 1974).

56. Lund, J.W., "Stability and Damped Critical Speeds of a Rotor in Fluid-Film Bearings," ASME Paper 73-DET-103, Des. Engr. Tech. Conf., Cincinnati, OH (Sept 1973).

57. Newkirk, B.L. and Lewis, W., "Oil Film Whirl, An Investigation of Disturbances due to Oil Films in Journal Bearings," IUTAM Symp. Dyn. Rotors, Lyngby, Denmark (Aug 12-16, 1974).

58. Duckworth, S.J. and Rieger, N.F., "Computerization of Threshold Speed Analysis in Flexible Rotor-Bearing Systems," Union College Ind. Study Rept. (Oct 1970).

59. Pinkus, O.J., "Experimental Investigations of Resonant Whip," Trans. ASME, 65, p 78 (Jan 1956).

60. Gunter, E.J., "Dynamic Stability of Rotor-Bearing Systems," NASA Rept. SP-113 (1966).

61. Gunter, E.J., Proprietary Computer Program, Dept. Mech. Engrg., Univ. Virginia, Charlottesville (1969).

62. Jennings, U.D. and Ocvirk, F.W., "The Simulation of Bearing Whirl on an Electronic-Analog Computer," J. Engr. Indus., Trans. ASME, Series D, pp 503-510 (1962).

63. Tondl, A., "Some Problems of Rotor Dynamics," Publishing House, Czechoslovak Acad. Sci., Prague (1965).

64. Tondl, A., "Self-Excited Vibrations," Natl. Res. Inst. Mach. Des. Bechovice, Monographs and Memoranda No. 9 (1970).

65. Tondl, A., "Resonance Vibrations of Non-Linear Systems Excited by a Periodic Non-Harmonic Force," Natl. Res. Inst. Mach. Des. Bechovice, Monographs and Memoranda No. 13 (1972).

66. Tondl, A., "Some Problems of Self-Excited Vibration of Rotors," Natl. Res. Inst. Mach. Des. Bechovice, Monographs and Memoranda No. 17 (1974).

67. Ker Wilson, W., Practical Solution of Torsional Vibration Problems, Chapman Hall, London (1963).

68. Nestorides, E.J., A Handbook on Torsional Vibrations, Cambridge Univ. Press (1958).

69. Bishop, R.E.D., "A Note on a Shaft Failure Induced by Gear Excited Vibrations," J. Mech. Engr. Sci. (Nov 1962).

70. Rieger, N.F., "Influence of Gear Errors on Drive Train Vibrations," Mach. Des., pp 115-119 (July 10, 1969).

71. Laskin, I., Proprietary Computer Program, Mechanical Technology Inc., Latham, NY (1969) See also Badgley, R.H. and Laskin, I., "Program for Helicopter Gearbox Noise Prediction and Reduction," USAAVLABS Tech. Rept. 70-12 (Mar 1970).

72. Laskin, I., "Prediction of Gear Noise from Design Data," Amer. Gear Manu. Assoc. Mtg., Clayton, MO, AGMA 299.03 (Oct 1968). See also Laskin, I., Orcutt, F.K., and Shipley, E.E., "Analysis of Noise Generated by UH-1 Helicopter Transmission," USAAVLABS Tech. Rept. 68-41 (June 1968).

73. Schneider, E., "Mechanical Problems Associated with Rolling-Mill Motors," Brown Boveri Rev., 51 (7), p 412 (1964).

74. Rieger, N.F. and Smalley, A.J., "Dynamic Loads on Gear Teeth From Transient Shock," Paper presented at AGMA Mtg., St. Louis, MO (Oct 25-28, 1970).

75. Kuszaj, A.M., Voelker, F.C., and Smalley, A.J., "Measurements and Analysis of Dynamic Torques Resulting from Shock Loading in a Large Rolling Mill," J. Engr. Indus., Trans. ASME, Vib. Conf., Toronto (1971).

76. Hizume, A., "Transient Torsional Vibration of Steam Turbine and Generator Shafts due to High Speed Reclosing of Electric Power Lines," ASME Paper No. 75-DET-71 (May 1975).

77. Ewins, D. J., "Vibration Characteristics of Bladed Disk Assemblies," J. Mech. Engr. Sci., 15 (3) (1973).

78. Cottney, D.J. and Ewins, D.J., "Towards the Efficient Vibration Analysis of Shrouded Bladed Disk Assemblies," J. Engr. Indus., Trans. ASME, p 1054 (Aug 1974). Paper No. 73-DET-144.

79. Cottney, D.J. and Ewins, D.J., "On Predicting the Natural Frequencies of Shrouded Bladed Disks," ASME Vib. Conf., ASME Paper No. 75-DET-113 (Sept 1975).

80. Ellington, J.P. and McCallion, H., "Blade Frequency of Turbines Effect of Disk Elasticity," Engineering Mag., p 645 (May 15, 1959).

81. Johnson, D.C. and Bishop, R.E.D., "The Modes of Vibration of a Certain System Having a Number of Equal Frequencies," *J. Appl. Mech.*, Trans. AMSE, p 379, Paper No. 56-APM-16 (Sept 1956).

LITERATURE REVIEW

survey and analysis of the Shock and Vibration literature

The monthly Literature Review, a subjective critique and summary of the literature, consists of two to four review articles each month, 3,000 to 4,000 words in length. The purpose of this section is to present a "digest" of literature over a period of three years. Planned by the Technical Editor, this section provides the DIGEST reader with up-to-date insights into current technology in more than 150 topic areas. Review articles include technical information from articles, reports, and unpublished proceedings. Each article also contains a minor tutorial of the technical area under discussion, a survey and evaluation of the new literature, and recommendations. Review articles are written by experts in the shock and vibration field.

In this issue of the DIGEST, review articles on ship hull vibration and underwater fluid-structure interaction are continued.

Dr. Chen continues his review of Underwater Fluid-Structure Interaction with an article on Mechanically-Applied Forces. The second review article deals with modeling physical phenomena involved in ship hull vibration.

UNDERWATER FLUID-STRUCTURE INTERACTION PART II: MECHANICALLY-APPLIED FORCES

L.H. Chen* and M. Pierucci**

Abstract - Digital computers are now commonly used in the dynamic analyses of complex structures in vacuo. Several general-purpose computer programs using the finite-element method are available; e.g., the NASTRAN codes developed by the NASA Langley Research Center. Specialized programs, such as the BOSOR code developed at the Lockheed Research Center (which is applicable for axi-symmetric shells including branched shell capability) have increased the computational efficiency of and expanded the use of computers in practical design and analysis. This capability has now been extended to the study of a structure subjected to one or more periodic forces and immersed in a stationary, inviscid-fluid medium. The problem is represented by equations (4) to (7). (see the April 1977 issue of the Shock and Vibration Digest.)

SOUND RADIATION

The solution of fluid-structure interaction problems involves determination of a relationship between the interaction pressure p and the normal velocity u. With formal methods, such a relationship satisfies the reduced wave equation, equation (5), and the velocity and pressure gradient relationship of equation (7). The formal methods considered below include integral methods and fluid finite-element methods.

Integral Methods

Integral methods are particularly suited for unbounded fluid problems in which the interaction involves only the fluid-structure interface. Formulation of a model for the infinite fluid using finite elements or finite differences is thus avoided. Two methods that have been reported in the literature include the simple-source method and the Helmholtz-formulation method.

In the simple-source method, the p-u relationship is represented by equations of the form

$$\begin{aligned} u(x) &= 2\pi\,\sigma(x) + \int_S \sigma(y)\,\frac{\partial G(x,y)}{\partial n}\,ds \\ p(x) &= -i\omega\rho \int_S \sigma(y)\;G(x,y)\,ds \end{aligned} \tag{13}$$

where $G(x, y) = \frac{e^{-ikr}}{r}$ is the free space Green's function; r is the distance between x and y. In the equation x denotes any point on the closed surface S (fluid-structure interface); y is a dummy variable for x in the integrands. The source-density function σ is an unknown relating p and u. The discrete version of equation (13) may be written as a pair of matrix equations:

$$\begin{aligned} u(x) &= X\,\sigma(y) \\ p(x) &= H\,\sigma(y) \end{aligned} \tag{14}$$

The elements of the matrices X and H are surface integrals involving the integrands $\frac{\partial G}{\partial n}$ and G respectively. The surface integrals depend on the geometry of the surface S and the wave number k; they are independent of the interaction problem and can be evaluated numerically with digital computers. Note that the matrix

$$Z = HX^{-1} \tag{15}$$

represents the specific radiation impedance as usually defined in the literature.

In the Helmholtz-formulation method, the p-u relationship is represented by

$$\begin{aligned} p(x) = {} & \frac{1}{2\pi}\int_S p(y)\cdot\frac{\partial G(x,y)}{\partial n}\,ds \\ & + \frac{i\omega\rho}{2\pi}\int_S u(y)\cdot G(x,y)\,ds \end{aligned} \tag{16}$$

or, in its discrete version, by

$$p(x) = \lambda\,p(y) + \gamma\,u(y) \tag{17}$$

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The elements of the matrices λ and γ are essentially the same type of surface integrals, involving the free-space Green's function and its gradient, as those of the matrices X and H. The matrices λ and γ do differ, however, in some numerical details; e.g., definition of the surface normal n.

In theory, the simple-source method can be thought of as a special or degenerate case of the Helmholtz formulation because the sum or difference of the exterior and interior Helmholtz formulations is used. (The simple-source method can be applied to either the exterior or the interior of a closed surface S.) In practice, the computational time required for both methods is reasonable if the surface is rigid. The Helmholtz method generally requires more than twice the computational time as the simple-source method for a fluid-structure interaction problem, primarily because a greater number of matrix inversions and multiplications (such as the γ matrix) are required to formulate the interaction equations. In addition, field-point quantities must be obtained after the equations have been solved.

The theory of integral methods is not always applicable. In such cases the identity of either p or u on a closed surface S will not allow determination of the other. These exceptional cases involve eigenvalues of the wave number k. These eigenvalues permit nonzero solutions of the reduced wave equation inside S if p or u on S is zero. These cases have been referred to as operator *singularities*, *inside eigenvalues*, and *cavity* resonance. One way to avoid such cases is to use the Helmholtz formulation to force a zero field inside S; the result is an overdetermined system of equations that can be solved.

A thorough review of recent work in using integral methods, including a summary of available computer programs, has been published [31]. A series of computer programs using the simple-source method in conjunction with the finite-element representation of structures dates from the early 1960s [15]. These computer programs have been used to study a number of shell-fluid interaction problems. In the late 1960s ill-conditioned matrices and unreasonable results associated with the cavity resonance problem were encountered during the analysis of a sonar array problem. One of the schemes now available to avoid the cavity resonance problem is to use the Helmholtz formulation to force a zero field inside the closed surface [3]. An improved method, in which the overdetermined system of equations is solved using the least square method has been developed [47]. The Helmholtz-formualtion and an overdetermined system of equations are seldom necessary; the *simple-source method* is usually used to solve fluid-structure interaction problems, probably because most of the practical structures for which complete interaction of a closed surface (i.e., a finite shell) must be studied are amenable to analysis only in the low frequency range.

At higher frequencies, where cavity resonances are likely to occur, a finite shell can generally be approximated by an infinite cylinder or plate for the purpose of sound radiation analysis. Integral methods are thus a powerful tool for computer analysis of sound radiation problems and can be made compatible with most currently used structural analytical techniques.

Fluid Finite-elements

The use of fluid finite elements to represent structures in analyses of sound radiation, scattering, and shock response is a relatively recent development. Either the fluid pressure element, in which the pressure in the fluid and the displacements in the structure are unknowns, or the fluid displacement element, in which the displacements in both fluid and structure are unknown, are used.

In theory, fluid pressure elements satisfy the equation *of equilibrium at the fluid-structure interface*. Pressure is the unknown at each nodal point in the fluid field. The set of equations, however, involves a mixture of unknown fluid pressures and unknown structural displacements and lacks the desirable matrix properties usually encountered in the finite element analysis of structures. The fluid displacement element scheme satisfies the condition of compatibility -- that is, continuous displacement or velocity -- at the fluid-structure interface. The three unknowns at each node in the fluid field are the three components of displacement (in a three-dimensional problem). The coefficient matrices of the equations are symmetric and banded.

Few papers have been published on the application of fluid finite elements to the study of sound and shock in fluid-structure interaction [48, 49]. A recent computer program [2] uses a fluid pressure element formulated in 1969 [58] and a piezoelectric solid

element (for sonar transducers) formulated in 1970 [1]. In other work a dummy displacement vector is used [20] to implement the fluid pressure element; NASTRAN capability is retained, and no computer code for the fluid pressure element must be written. A fluid displacement element approach has also been described [31]. More computational experiences are necessary before pressure and displacement fluid elements can be evaluated. In theory the two approaches differ in the boundary connecting equations at the fluid-structure interface and in the fluid field, perhaps providing the upper-bound and lower-bound solutions as typical of problems in the classical theory of elasticity in which either the compatibility or the equilibrium condition is not explicitly met. When both approaches were applied to the problem of a vibrating sphere, each gave satisfactory results when compared to the available exact solution. The sphere problem is a relatively simple one, however, so that the results cannot be compared as to accuracy and computer time. The application of the fluid finite element to a bounded fluid region is direct, but special techniques are necessary for problems involving infinite fluid boundaries [31].

Special Methods

For the solution of steady-state sound radiation and scattering problems, no general formal method is available other than the integral and finite element methods. The classical method of separation of variables can be used only when the vibrating surface can be represented by a suitable coordinate system; the functions can be tabulated or calculated with computers. Such analytical results also often allow better physical insight into the problem. Early analytical work for cylindrical shells dates from the 1950s [7, 30, 55]. More recent work on spherical and spheroidal shells has been reviewed [19].

One practical approach to formulating a general computer program for engineering design and analysis involves the matrix method [14]. The interaction problem was formulated so that calculations of the structural and acoustic components were separate from the interaction equation. The structure is characterized by its dynamic properties in vacuo (structural mobilities). The acoustic field is described in terms of specific radiation impedances; these are functions of the geometry of the interface surface but are independent of its structural or elastic properties. In such a formulation, either the structural mobilities, the acoustic impedances, or both may be obtained analytically or experimentally. This approach utilizes the analytic results of acoustic radiation impedances that are available in the literature. An example of this approach is a computer program in which NASTRAN capability is extended to fluid-structure interaction with the integral formulation for the acoustic field [27]. Calculations of structural and acoustic properties are completely independent, however, so as to provide a general link to the utilization of acoustic results.

STRUCTURAL VIBRATIONS

The formal methods described for sound radiation are of course applicable to the study of vibratory motion of a structure. The case of fluid-filled shells has been reviewed [17]; analytical work on spherical, spheroidal, and circular cylindrical geometry, including both steady-state and transient responses, is included.

This section deals briefly with decoupling methods, the added or virtual mass method has been used for many years to study the dynamic response of ship structures. If the fluid medium is incompressible (i.e., ρ is constant, giving $c \to \infty$), the wave number k vanishes and the reduced wave equation, equation (5), reduces to the Laplace equation. With a principal oscillation, the only effect of the fluid medium on the structure is an increase in the generalized mass of the structure in vacuo by the added mass of the fluid. A mathematical equivalent of an incompressible fluid is the case of a finite acoustic speed c at lower frequencies ($\omega \to 0$); the wave number k also vanishes in this case.

It is instructive to look at the added mass approximation in terms of acoustic radiation impedance, Rewrite the frequency-dependent radiation impedance matrix $Z(\omega)$ of equation (15) as follows.

$$Z(\omega) = [Z_{jk}] \tag{18}$$

The matrix element Z_{jk} is called the specific radiation impedance; it denotes the pressure at location j due to a unit velocity at location k. The cases for $j = k$ and $j \neq k$ are usually referred to as self and mutual impedances, respectively. The total pressure p_j at each j is related to the velocities by equation (19).

$$\{p_j\} = [Z_{jk}] \; \{u_k\} \tag{19}$$

The symbol $\{\ \}$ denotes a column matrix or vector; for periodic time dependency, equation (19) becomes

$$\{p_j\} = \frac{1}{-i\omega}[Z_{jk}] \quad \{u_k^*\} \qquad (20)$$

where u_k^* denotes acceleration. The specific radiation impedance can be written in terms of its real and imaginary parts.

$$Z_{jk} = R_{jk} + iI_{jk} \qquad (21)$$

The added mass approximation, also called the incompressible fluid approximation, involves two approximations of the exact expression of equation (21). The first is that the real part of the specific radiation impedance is negligible; the second is that the ratio of the imaginary part to the frequency is independent of frequency at lower frequencies. A matrix of added mass coefficients is obtained when these approximations are applied to equations (20) and (21).

$$[m_{jk}] = [I_{jk}]/\omega = \text{const.}\ (\omega \to o) \qquad (22)$$

Note that the matrix $[m_{jk}]$ is in general a full matrix and can be obtained from the imaginary part of the specific acoustic radiation impedance at an arbitrarily chosen low frequency value ($\omega \to o$).

The pressure at mass j is given by equation (23).

$$p_j = \sum_k m_{jk} u_k^* = \sum_k \left(\frac{I_{jk}}{\omega}\right)_{\omega \to o} u_k^* \qquad (23)$$

An added mass M_j can be defined by equation (24).

$$M_j = \sum_k m_{jk} \frac{u_k^*}{u_j^*} \qquad (24)$$

It is important to point out that the added mass coefficients m_{jk} are pure fluid field quantities; that is, they are independent of frequency ω at $\omega \to o$ and are a function only of the geometry of the vibrating surface. The added mass M_j, however, because of u_j^* and u_k^* also depends on the dynamic properties of the structure; for example, natural frequencies and mode shapes of free vibration of the structure in vacuo. It is in this context that added mass is often spoken of as an important function of frequency.

In ship vibration, the first methods for calculating added mass for two-dimensional hull forms resembling the sections of many types of ships (except bulbous bows, multi-hulls, or shapes with some discontinuity) are known as Lewis forms. Many added mass methods, including some modified Lewis form methods, are now used by ship designers. In general, frequency-independent added mass is used for such high frequency vibration as propeller-excited hull vibration; frequency-dependent added mass is used for such rigid-body ship motions as heave and pitch [32, 44].

The major difficulty in calculating added mass for ship vibration studies is that hull shape cannot accurately be accounted for. Integral methods and fluid finite-element methods capable of dealing with an arbitrary three-dimensional shape can be utilized.

REFERENCES

1. Allik, H. and Hughes, T.J.R., "Finite Element Method for Piezoelectric Vibration," Intl. J. Numer. Methods Engr., 2, pp 151-157 (1970).

2. Allik, H., Cacciatore, P.J., Gauthier, R., and Gordon, S.F., "MARTSAM IV - A Version of GENSAM," General Dynamics, Electric Boat Division Report U440-74-043, Vols. I, II, III, IV (May 1974). (approval of NAVSEA required.)

3. Baylor, J.L., "Operator Singularities of Integral Equations of Linear Acoustics," J. Acoust. Soc. Amer., 42 (5), p 1204 (Nov 1967).

4. Belsheim, R.O. and O'Hara, G.J., "Shock Design of Shipboard Equipment," Part I (unclassified NAVSHIPS Rept. No. 250-423-30 (May 1961)).

5. Benjamin. B.T., "Effect of a Flexible Boundary on Hydrodynamic Stability," J. Fluid Mech., 9, pp 513-532 (1960).

6. Berger, B.S., "Reflected and Radiated Acoustic Pressure from an Elastic Shell of Revolution," Rept. Contract No. N00014-67-A0239-0020, Univ. Maryland (Oct 1975).

7. Bleich, H.H. and Baron, M.L., "Free and Forced Vibrations of an Infinitely Long Cylindrical Shell in an Infinite Acoustic Medium," J. Appl. Mech., Trans. ASME, 21, pp 167-177 (1954).

8. Bleich, H.H., DiMaggio, F.L., and Baron, M.L., "On Uncoupling Fluid-Structure Interaction Problems," Part II, Weidlinger Assoc. Rept. ONR Contract N00014-72-C-0119 (July 1973).

9. Blick, E.F., "Skin Friction Drag Reduction by Compliant Coatings," Paper F2, Intl. Conf. Drag Reduction, Cambridge (Sept 1974).

10. Boggs, F.W. and Frey, H.R., "The Effect of a Lamiflo Coating on a Small Planing Hull Having Zero Deadrise," U.S. Rubber Co. Rept. (June 1961).

11. Breslin, J.P., "Techniques for Estimating Vibratory Forces Generated by Propellers," Tech. Res. Bull. No. 1-34, Hydrodynamics Committee, Soc. Naval Architects Marine Engr. (1975).

12. Brown, R., "Turbulent Pressure Spectrum Measurements on a Compliant Surface," Fourth Biannual Symp. Turbulence in Liquids, Univ. Missouri-Rolla (Sept 1975).

13. Carrier, G.F., "The Interaction of an Acoustic Wave and an Elastic Cylindrical Shell," Tech. Rept. 4, Contract Nonr-35810, Brown Univ. (1951).

14. Chen, L.H., "A Matrix Method of Analysis of Structure-Fluid Interaction Problems," ASME Paper No. 61-WA-220 (Nov 1961).

15. Chen, L.H. and Schweikert, D.G., "Sound Radiation from an Arbitrary Body," J. Acoust. Soc. Amer., 35 (10), pp 1626-1632 (1963).

16. Davies, H.G., "Sound From Turbulent Boundary Layer Excited Panels," J. Acoust. Soc. Amer., 49 (3), pp 878-889 (1971).

17. DiMaggio, F.L., "Dynamic Response of Fluid-Filled Shells," Shock Vib. Dig., 7 (5), pp 5-12 (May 1975).

18. Dowell, E.H., "Noise or Flutter or Both?" J. Sound Vib., 11 (2), pp 159-180 (1970).

19. Engin, A.E. and Engin, A.W., "Survey of the Dynamic Response of Spherical and Spheroidal Shells," Shock Vib. Dig., 7 (3) (1975).

20. Everstine, G.C., Schroeder, E.M., and Marcus, M.S., "The Dynamic Analysis of Submerged Structures," Fourth NASTRAN Users Colloq., Langley Res. Ctr. (Sept 1975).

21. Farn, C.L.S. and Huang, H., "Transient Acoustic Fields Generated by a Body of Arbitrary Shape," J. Acoust. Soc. Amer., 43 (2), pp 252-257 (1968).

22. Feit, D., "Flow Noise Characteristics of Elastic Plates Excited by Boundary Layer Turbulence," Cambridge Acoust. Assoc. Rept. No. U-279-199 (Apr 1968).

23. Ffowcs-Williams, J.E., "Reynold Stress near a Flexible Surface Responding to Unsteady Air Flow," Bolt Beranek and Newman Rept. No. 1138 (June 1964).

24. Geers, T.L., "Residual Potential and Approximate Methods for Three Dimensional Fluid-Structure Interaction Problems," J. Acoust. Soc. Amer., 49 (5), pp 1505-1510 (1971).

25. Hansen, R.J. and Hunston, D.L., "An Experimental Study of Turbulent Flows Over Compliant Surfaces," J. Sound Vib., 34 (3), pp 297-308 (1974).

26. Haviland, J.K. and Cooley, D.E., Structural Mechanics Computer Programs (W. Pilkey, K. Saezalski, and H. Schaeffer, Eds.), Univ. Virginia Press (1974).

27. Henderson, F.M., "A Structure-Field Interaction Capability for the NASA Structural Analysis (NASTRAN) Computer Program," DTNSRDC Rept. No. 3962 (Aug 1972).

28. Huang, H. and Wang, Y.F., "Early-Time Interaction of Spherical Acoustic Waves and a Cylindrical Elastic Shell," J. Acoust. Soc. Amer., 50, pp 885-891 (1971).

29. Izzo, A.J., "An Experimental Investigation of the Turbulent Characteristics of a Boundary Layer Flow over a Vibrating Plate," Dissertation submitted to Univ. Connecticut (1969).

30. Junger, M.C., "Vibrations of Elastic Shells in a Fluid Medium and the Associated Radiation of Sound," J. Appl. Mech., Trans. ASME, 19 (1) (Mar 1952).

31. Kalinowski, A.J., "Fluid Structure Interaction," in Shock and Vibration Computer Programs - Reviews and Summaries (W. and B. Pilkey, Eds.), Naval Res. Lab., Wash., D.C., pp 405-452 (1975).

32. Kline, R.G. and Daidola, J.C., "Ship Vibration Prediction Methods and Evaluation of Influence of Hull Stiffness Variation on Vibratory Response," Rept. SSC-249, Ship Struc. Comm., U.S. Coast Guard, Wash., D.C., pp 2-11 (1975).

33. Klosner, J.M., "Response of Shells to Acoustic Shocks," Shock Vib. Dig., 8 (5), pp 3-13 (1976).

34. Krämer, M.O., "Boundary Layer Stabilization by Distributed Damping," J. Amer. Soc. Nav. Engr., 72, p 25 (1960).

35. Landhal, M.T., "On the Stability of a Laminar Incompressible Boundary Layer over a Flexible Surface," J. Fluid Mech., 13 (1962).

36. Laufer, J. and Maestrello, L., "The Turbulent Boundary Layer over a Flexible Surface," The Boeing Co. Document No. D6-9708 (1963).

37. Leibowitz, R.D., "Vibroacoustic Response of Turbulence Excited Thin Rectangular Finite Plates in Heavy and Light Fluid Media," J. Sound Vib., 40 (4), pp 441-495 (1975).

38. Leibowitz, R.C., "Experimental and Theoretical Foundations for Modern Turbulence - Vibroacoustic Research. A Decade of Exploration 1956-1965," DTNSRDC Tech. Note SAD 536-194 (Nov 1975).

39. Lin, C.C., The Theory of Hydrodynamic Stability, Cambridge Univ. Press (1967).

40. McMahon, J.F., "Surface Wave Characteristics for the Flow of an Inviscid Fluid Over an Elastic Surface," Penn. State Univ., ORL Rept. No. TN502-02 (1970).

41. Menton, R.T. and Magrab, E.B., "Interaction of Acoustic Pulses with Fluid-Loaded Shell Structures," Shock Vib. Dig., 5 (12), pp 2-12 (1973).

42. Mnev, Y.N. and Pertsev, A.K., "Hydroelasticity of Shells," Air Force Systems Command Rept. FTD-MT-24-119-71 (1971).

43. Morfey, C.L., "Rotating Blades and Aerodynamic Sound," J. Sound Vib., 28 (3), pp 587-618 (1973).

44. Neubert, V.H., "Inertia Matrices for Finite Elements," Shock and Vibration Computer Programs - Reviews and Summaries (W. and B. Pilkey, Eds.), Naval Res. Lab., Wash., D.C., pp 625-645 (1975).

45. Pierucci, M., "Hydrodynamic Instabilities of a Layered Medium in the Presence of a Flow Field," J. Acoust. Soc. Amer., 58, p 521 (1975).

46. Ross, D., "Vortex Shedding Sounds of Propellers," Bolt Beranek and Newman Rept. No. 1115 (1964).

47. Schenck, H.A., "Improved Integral Formulation for Acoustic Radiation Problems," J. Acoust. Soc. Amer., 44 (1), pp 41-58 (July 1968).

48. Smith, R.R., Hunt, J.T., and Barach, D., "Finite Element Analysis of Acoustically Radiating Structures with Applications to Sonar Transducers," J. Acoust. Soc. Amer., 54 (5), pp 1277-1288 (1973).

49. Smith, R.R., Barach, D., McCleary, L.E., and Johnson, J.L., "A Finite Element Model for Analyzing Self-Noise Levels in Geophysical Towed Line Arrays," ASME Paper No. 76-PVP-27 (May 1976).

50. Strawderman, W.A., "Turbulence Induced Plate Vibrations: An Evaluation of Finite and Infinite Plate Models," J. Acoust. Soc. Amer., 46 (5), pp 1294-1307 (1969).

51. Tokita, N. and Boggs, F.W., "Final Report on Theoretical Study of Compliant Coatings to Achieve Drag Reduction on Underwater Vehicles," U.S. Rubber Co., Rept. No. NR062-241 (1962).

52. Tsakonas, S., Jacobs, W.R., and Ali, M.R., "An Exact Linear-Surface Theory for a Marine Propeller in a Non-Uniform Flow Field," J. Ship Res. (Dec 1973).

53. VonWinkle, W.A., "An Evaluation of a Boundary Layer Stabilization Coating," Underwater Sound Lab Memo No. 922-111-61 (1961).

54. Vorus, W.S., "An Integrated Approach to the Determination of Propeller-Generated Vibratory Forces Acting on a Ship Hull," Univ. Michigan Rept. No. 072 (1971).

55. Warburton, G.B., "Vibration of a Cylindrical Shell in an Acoustic Medium," J. Mech. Engr. Sci., 3, p 69 (1961).

56. White, F.M. and Quaglieri, R.E., "A Theoretical Estimate of Turbulent Wall Pressure Fluctuations on a Compliant Boundary," U.S. Navy Underwater Sound Lab. Rept. No. 767 (1966).

57. Zamyshiyayev, V.B. and Yakovelev, Y.S., "Dynamic Loads in Underwater Explosion," Rept. AD-757 183, NTIS, U.S. Dept. of Commerce, Springfield, VA (Feb 1973).

58. Zienkiewicz, O.C. and Newton, R.E., "Coupled Vibrations of a Structure Submerged in a Compressible Fluid," Proc. Intl. Symp. Finite Element Tech., Intl. Assoc. Ship Struc., Univ. Stuttgart, Ger. (June 1969).

A REVIEW OF SHIP HULL VIBRATION
PART II: MODELING PHYSICAL PHENOMENA

J. Juncher Jensen and Niels Fl. Madsen*

Abstract - Formulations of the equations of motion were described in Part I: Mathematical Models (see the April 1977, issue of The Shock and Vibration Digest). In order to solve these equations a number of physical quantities that describe the ship and the surrounding water must be determined. These quantities include hull stiffness, hull mass and virtual added mass of water, exciting forces, and damping. The accuracy to which these quantitites must be known depends on the complexity of the mathematical model and the distributions of the physical quantities. The value of some quantities depends on the frequency and mode of vibration.

HULL STIFFNESS

For simple beam models hull stiffness is usually characterized by cross-sectional rigidities in bending, shear, and torsion. In principle, all rigidities, as well as the position of the neutral axis, can be determined from the dimensions of the ship's frame. The thin-walls of the hull complicate the problem, however. Calculations of bending ridigities must account for the shear lag effect, especially in the higher vibration modes. A fundamental treatment of the shear lag effect as it relates to the design of ship structures has been published [110, 111].

The shear coefficient k in the expression for shear rigidity in equation (1) accounts for the variation in shear stresses across the cross section. Several methods have been used to determine k. The approach suggested by Cowper [16] seems to yield the best results [101]. With this method the three-dimensional equation of elasticity is reduced to the Timoshenko beam equation. Other methods currently in use tend to give larger values of k [101], but are widely used because they are simple [3, 17, 48, 96]. Bredt's formula is usually used to find the St. Venant torsional rigidity (GK) warping rigidities are determined with semi-empirical formulas [48]. Piecewise uniform cross sections are normally assumed [48, 96] for locating the shear center. Evaluation of stiffness properties in three-dimensional finite element analysis is straightforward but cumbersome.

HULL MASS AND VIRTUAL ADDED MASS OF WATER

The masses in the equations of motion include the weight of the ship, the deadweight (pay load), and the virtual added mass of water. Distribution of the weight of the ship can be calculated from drawings. Calculations of the distribution of the deadweight must include some consideration of the effect of the load [64]. The influence of the hydrodynamic pressure on the submerged surface of the vibrating hull can be accounted for in the equations of motion [73] with a term called the added mass (virtual) or the entrained water.

Lewis [82] did the fundamental work on the added mass concept in ship vibration. In his strip theory approach, he divided the hull into sections with uniform cross sections. He also assumed that the water flow around the sections is two-dimensional and that the cross section of the hull is symmetric about the water plane.

The analytical solution to the problem of two-dimensional flow around the hull involves two-parameter mapping. The family of cross sections (Lewis forms) resulting from two-parameter mapping closely resembles normal hull shapes.

$$\frac{z}{b_0} = \zeta + \frac{a_1}{\zeta} + \frac{a_3}{\zeta^3} \tag{3}$$

In the mapping, equation (3), z and ζ are complex coordinates of the Lewis forms and the unit circle, respectively: b_0 is a scale factor of dimension length. The two real coefficients a_1 and a_3 and the scale factor b_0 can be expressed in terms of the ratio λ between the draught d and the half-beam b and in terms of the sectional area coefficient $c = S/2bd$.

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S is the immersed area of the section [57]

$$a_1 = \frac{b}{2b_0}(1-\lambda) \quad a_3 = \frac{b}{2b_0}(1+\lambda) - 1$$

$$\frac{b_0}{b} = \frac{1}{4}\left[3(1+\lambda) - \sqrt{(1+\lambda)^2 + 8\lambda(1-4c/\pi)}\right] \tag{4}$$

Lewis used analytical results [73] for a circular cross section to derive the two-dimensional kinetic energy of the entrained water. He used the conformal mapping technique, equation (3), to calculate the added mass per unit length for a Lewis form cross section. The added mass per unit length for vertical motions is shown in equation (5).

$$\mu_V = \frac{\pi}{2}\rho b_0^2\left[(1+a_1)^2 + 3a_3^2\right] \tag{5}$$

The solutions for horizontal and rotatory motions of a Lewis form cross section are similar; the added mass per unit length for the horizontal motion [74] is shown in equation (6).

$$\mu_h = \frac{2}{\pi}\rho b_0^2\left[\left(\frac{d}{b_0}\right)^2 + \frac{16}{3}a_3^2\right] \tag{6}$$

The added mass moment of inertia per unit length for rotation about the intersection line between the lateral plane and the waterline plane [57] is shown in equation (7).

$$J_\mu = \pi\rho b_0^4\left[a_1^2(1+a_3)^2 + 2a_3^2\right] \tag{7}$$

Kumai [57] has also derived the added mass moment of inertia for rotation about an arbitrary line in the lateral plane.

Other two-parameter mappings give better approximations of sections near the stern [105]. A three-parameter mapping has been proposed in which the third geometrical quantity is the moment of inertia of the immersed cross section [75]. Tasai [123] and Porter [103] considered the generalized map shown in equation (8).

$$\frac{z}{b_0} = \zeta + \sum_{i=1}^{n} a_{2i-1}\zeta^{-(2i-1)} \tag{8}$$

Another general conformal map has been suggested [77], as has an alternative approach -- the source-sink method [19]. It should be noted that the amount of computation increases as the number of free parameters increases; it is for this reason that the Lewis form is still preferred in most calculations of ship hull vibration.

Three-dimensional effects must be considered before the virtual added masses calculated by any two-dimensional method is used. Lewis [82] and Taylor [124] defined a three-dimensional reduction factor J as the ratio between the kinetic energy of the exact three-dimensional flow and the kinetic energy of the approximate two-dimensional flow; both flows were calculated for the whole ship. The two-dimensional added masses or mass moments of inertia are multiplied by the corresponding reduction factor J to obtain the approximate values of the three-dimensional added masses and mass moments per unit length. Two approaches have been used to determine approximate values of the reduction factor; one is based on ellipsoidal bodies and the second on a cylinder.

In the first method [30, 52, 76, 82, 118, 124] the reduction factors are based upon exact three-dimensional solutions for ellipsoidal rigid bodies having two or three different main axes with the same dimensions as the hull. No theoretical solutions exist for elastic ellipsoidal bodies, but Townsin [128] has proposed the following empirical formula for vertical vibrations.

$$J_n = 1.02 - 3\left(1.2 - \frac{1}{n}\right)\frac{B}{L} \tag{9}$$

In equation (9) n is the number of nodes, and B is the water-plane breadth amidship. Equation (9) has been verified experimentally for modes having as many as four nodes. This simple equation is widely used.

In the second method for the determination of J a cylinder of either infinite length [24, 46, 78, 81, 124] or finite length [58, 62, 67, 72] is considered. Various cross sections of the cylinder -- circular, elliptical, rectangular, and Lewis forms -- are treated by assuming a sinusoidal vibration mode along the longitudinal axis. Kumai [62, 72] used a cylinder of length L and a Lewis form cross section to derive the following expression for vertical motions.

$$J_n = \frac{16}{\pi^2\left[(1+a_1)^2 + 3a_3^2\right]}\sum_m \frac{m^2}{(m^2-n^2)^2}\left\{\frac{(1+a_1)^2}{1+k_m K_0(k_m)/K_1(k_m)} + \frac{9a_3^2}{3+k_m K_2(k_m)/K_3(k_m)}\right\} \tag{10}$$

In this equation $k_m = m\, b_0/L$, n is the number of nodes and $K_i(\ )$ is the i^{th} order modified Bessel function of the second kind. The remaining quantities are defined in equation (4). The summation in equation (10) is over-all even positive numbers of m if n is odd and vice versa. Measurements [12] showed close agreement for the two-node vibration mode [72]. Reduction factors, calculated according to equations (9) and (10), are shown in figure 1 for a 340,000 ton deadweight (tdw) tanker.

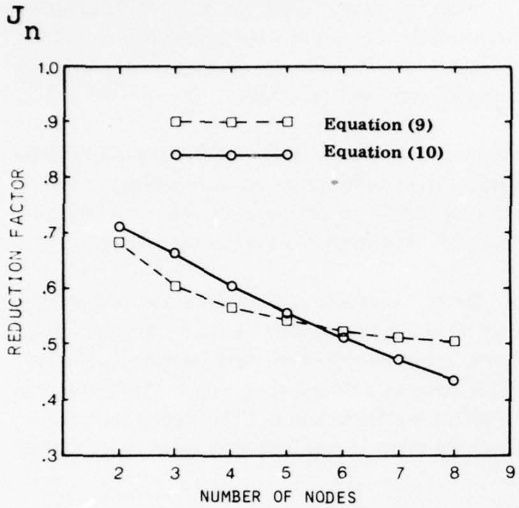

Figure 1. Three-dimensional reduction factor for the added water in vertical vibrations for a 340,000 tdw tanker.

The reduction factor J_n should vary along the ship hull. An empirical correction [3] has been used by Kumai [67]. Of particular interest among the approaches for evaluating three-dimensional added masses is a variational formulation. The Lagrangian method is used to couple the energy in both the elastic body and the surrounding fluid. An early application of this method was given by Csupor [17]. Landweber [78, 79] later applied the principle to the vibration of a beam with circular cross sections. Misra [93] recently considered a Timoshenko beam with arbitrary cross sections vibrating vertically in an ideal fluid. Computed natural frequencies for ellipsoidal bodies were in close agreement with experimental results in most cases [128].

These methods assume that the cross sections do not deform; this is true, however, only for the lower modes of vibration. The effect of the distortion on the added mass in the two-beam model [47, 97] and the beam-plate model [98] is considered by assuming that the elastic deflection of the bottom of the hull is parabolic in the athwartships direction. The two-dimensional added masses associated with the various parts of the models are calculated from the kinetic energy of the two-dimensional flow around the deformed cross section. A coupling term also appears.

A three-dimensional finite element structural model [28] requires a three-dimensional finite element formulation of the fluid for consistent results. Fluid finite elements have been considered in some detail [14].

None of the idealizations of the water surrounding the ship considered the influence of ocean waves on vibration. It has been shown [63], however, that frequency modulation must occur in waves due to the variation of the added mass with respect to time. The influence of water depth on the added masses is usually taken into account by empirical formulas [105].

EXCITING FORCES

The main sources of hull vibrations are the propeller, which creates periodic forces acting on the aft end of the ship, and waves, which create random forces along the entire length of the ship. Engine-unbalance was at one time an important source of vibration but seems now to have less influence than the propeller and waves [106].

Propeller-induced Forces

The motion of the propeller in the non-uniform wake field creates hydrodynamic forces of two types, propeller forces and surface forces. The propeller forces arise from the pressure acting on the propeller blades and are transmitted through the propeller shaft and bearings to the hull. The surface forces, also called hull forces or pressure forces, arise from the hydrodynamic pressure on the stern.

Several methods are available for determining propeller-induced forces [8, 10, 22, 106]. These forces can be accurately calculated with the lifting-surface theory, if cavitation and other nonlinear effects can be neglected; in the theory, which was first applied to ship propellers by Sparenberg [121], the hull-induced velocity field in the propeller plane is assumed to be known. Fluid particles must remain in contact with the propeller blades; it follows, therefore, that velocity components from the hull-induced wake and normal to the blade surfaces should equal the corresponding components of the propeller-induced fluid velocities. This condition can be expressed as a three-dimensional integral equation. A successful numerical solution of this equation required ten years of intensive research [22, 119, 129-133]. The solution yields the pressure distribution and the lift on the blades. The resulting propeller forces and moments are found by integration.

The most important results with respect to excitation of hull vibrations are the two propeller force components F_y and F_z in the transverse horizontal and vertical direction. The other four resultants generally give rise only to local vibrations in the shaft and propulsion system [41]. The dominant harmonic terms in the force components F_y and F_z usually vary at a frequency equal to the blade frequency. The blade frequency is defined as the product of the number of blades N and the angular velocity Ω of the propeller. The harmonic components F_{yN} and F_{zN} can be written as shown in equation (11).

$$F_{yN} = \mathrm{Re}\left\{\frac{1}{2} N e^{iN\Omega t} \int_0^{r_0} [L_{N-1}(r) + L_{N+1}(r)] \sin\theta_p(r) dr\right\}$$

$$F_{zN} = \mathrm{Re}\left\{\frac{1}{2i} N e^{iN\Omega t} \int_0^{r_0} [L_{N-1}(r) - L_{N+1}(r)] \sin\theta_p(r) dr\right\} \tag{11}$$

In the equation t denotes time, r_0 is the radius of the propeller, $\theta_p(r)$ is the geometric pitch angle at radius r, and $L_n(r)$ is the lift on the blade at radius r due to the n^{th} circumferential harmonics of the wake. From equation (11) it follows that only two wake harmonics -- the $(N-1)^{th}$ and the $(N+1)^{th}$ -- are responsible for the main part of the propeller induced hull vibrations. Similar formulas can be derived for the higher harmonic components of F_y and F_z. Recent full-scale measurements [135] indicate that these higher harmonics should also be taken into account.

Few correlations have been made between measured and calculated propeller forces [10], although the lifting surface theory has been widely used. Kuiper [53] showed that computed results are usually in good agreement with experimental results.

In order to reduce computer time, Breslin [9] developed a method for using a desk calculator to determine the propeller forces. Some of the operators depend only on propeller geometry, however, and they must be determined by a three-dimensional lifting-surface calculation. An alternative approach, the lifting line theory, is adequate if qualitative results are sufficient [60, 136].

Because of the dynamic behavior of the shaft system -- including the elastically supported bearings -- bearing forces can be 10 to 20 times greater than propeller forces [37] if resonance occurs in the system.

The surface forces acting on the hull can, in principle, be found if the component of fluid velocity, including the propeller-generated part normal to the ship hull, is zero. In a different approach [134], the surface forces are found without calculating the pressure due to diffraction; one source of error is thus eliminated.

At present, however, the influences of the hull surface and the water plane on the propeller-induced pressure distribution are ignored. The resulting surface forces are then corrected by a "solid-boundary" factor, usually assigned a value of two, which is the theoretical value for the flat plate and a circular cylinder. Three-dimensional finite-element analysis of hull vibration requires calculation of the pressure distribution on the hull, rather than the net surface forces. A constant solid-boundary factor of two at all points on the hull surface is not satisfactory: full-scale measurements [21] indicate that the factor increases with the horizontal distance from the propeller. Cavitation should be taken into account when surface forces are calculated because it can increase these forces by a factor of two to ten [33, 37, 99, 100]. Results of measurements of surface forces on models [59, 66, 102] generally agree with theoretical values. When cavitation occurs, however, results with models might not be applicable to ships due to scale effects [100]. It has also been found [32] that it is not always possible to measure

propeller-induced surface forces with pressure transducers; the reason is the additional pressure fluctuations generated by the vibrating hull.

Most of the experimental work on propeller-induced forces with both models [83, 84] and ships [135] involves measuring net forces on the hull. Lewis [84], in considering the influence of the rudder-propeller clearance on the net forces, found that the clearance is extremely important; the total transverse force varies almost cyclically with clearance, and the amplitude of the variations is large.

Wave-induced Forces

Wave forces, and the vibrations they induce, should be treated as random processes [85]. The lowest natural frequency of most conventional ships is well above the harmonic components of the wave energy spectrum. These harmonic components have enough energy to excite excessive hull vibrations. As the size and speed of ships have increased, however, this situation has begun to change; that is, the value of the lowest natural frequency has been increasing. Measurements of continuous wave-excited hull vibrations, called "springing," have been made on ships [2, 4, 86].

The effects of rigid body heave and pitch motions can be neglected [23, 25] for large ships because the wave lengths of the wave harmonics that generate hull vibrations are small compared with the length of the ship. The exciting forces are calculated from the Froude-Kriloff hypothesis. The pitch motion of smaller ships can be an important source of hull vibrations, however; it is calculated by determining the hydrodynamic force as the time derivative of the momentum of the added mass at the fore end [70, 71]. A unified treatment of both effects has been proposed [6, 49]. The randomness of the sea is accounted for by using the strip theory approach to calculate the response due to all pertinent wave harmonics. The principles of linear superposition are then applied. Three problems with this approach are the uncertainty of the high-frequency range of the wave spectrum, lack of verification of validity for use of the strip theory, and the effect of hydrodynamic damping. Recent experiments [137] have indicated, however, that -- for wave lengths greater than half the length of the ship -- the strip theory is satisfactory.

DAMPING

Little damping of hull vibration occurs, especially at lower modes. The effect of damping on the natural frequencies is thus usually neglected. Accurate calculation of the amplitude of forced vibration, however, requires calculation of the distribution of damping throughout the hull, the cargo, and the surrounding water. The vibration of a hull in water comprises a very complex system. Damping is not only a material property, but also depends on the vibration mode, the excitation forces, and the structural configuration. The calculation of damping in a ship is based largely on measurements of the rate of decay of oscillation (logarithmic decrement) or equivalent quantities in similar ships. Full-scale measurements from slamming or mechanical excitation tests have been reported [1, 2, 29, 44, 95, 125].

Sources of Hull Damping

Current knowledge of damping of hull vibrations has been summarized [5, 108]. A primary source of this damping is material hysteresis in the steelwork. In the very low stress region ($\sigma < 2 \text{MN/m}^2$) the logarithmic decrement is approximately constant for mild steel. Energy loss, denoted by d, increases in an almost quadratic manner with stress amplitude.

$$d = C\sigma^n \qquad (12)$$

C is a material constant and $n = 2$. In the intermediate stress region ($2 < \sigma < 200$ MN/m^2) the exponent n lies between two and three; in the high stress region ($\sigma > 200$ MN/m^2), damping increases rapidly, becoming dependent on stress [5]. For a virgin steel plate the logarithmic decrement ranges from 6×10^{-5} to 1.6×10^{-4}. That of a welded structure is much higher, typically ranging from 9×10^{-3} to 3×10^{-2} [120], partly because high stresses result from stress concentrations and partly because of residual welding stresses.

Coulomb damping, or dry friction, in the hull is important in riveted structures. Johnson [132] performed vibration tests on sister ships; one was welded, the other was riveted. Damping for the riveted ship was almost twice that for the welded ship.

Energy dissipates throughout solid, dry cargos and bulk, liquid cargos. Measurements have shown that the cargo usually increases the logarithmic decrement by a small amount [1, 2, 44, 125]. Damping forces in the cargo are thus supposed to be as important as structural damping.

The least important source of damping appears to be that due to water. Sezawa and Watanabe [118] classified external ship damping according to source: water friction, generation of surface waves, and generation of pressure waves. Water friction results from form resistance and frictional resistance that depends on the extent of marine life attached to the hull. The effect of surface waves is negligible because of the very small vibration amplitude, and significant pressure waves (sound) are not generated at low frequencies.

Representation of Damping in Calculations

Kumai [56] concluded that the linear Voigt-type viscoelastic stress-strain relationship given by equation (13)

$$\sigma = (E + E^* \frac{\partial}{\partial t})\,\epsilon \tag{13}$$

is the most convenient mechanism for explaining internal damping. Furthermore, he [56] also found that external damping is negligible. The material damping factor E^* for steel has been found from model tests [80]. Equation (13) shows that a structure is more flexible at a low deformation rate and more rigid at a high deformation rate. Equation (1), given by Kumai [56] for a Timoshenko beam, includes the effects of external viscous damping and of visco-elasticity. Neglecting external damping, the logarithmic decrement for a uniform beam model becomes [65]

$$\delta_i = \frac{1 + (\frac{\nu}{\eta}\gamma + \beta)\, i^2 \pi^2}{1 + (\gamma + \beta)\, i^2 \pi^2}\ \pi\eta\omega_i \tag{14}$$

In Equation (14) $\gamma = EI/kGAL^2$, $\beta = r^2/L^2$ (r is the radius of gyration); η and ν are the normal and tangential damping factors. Kumai [70] suggested the values $\eta = 1.7 \times 10^{-3}$ sec and $\nu = 0.45 \times 10^{-3}$ sec; he assumed that the damping factor ν due to tangential viscosity equals that of torsional vibration and that the external damping factor equals the external damping factor found from model tests.

The viscoelastic assumption, equation (13), has been expanded to include the damping from cargo and surrounding water as a nonlinear external damping mechanism [95]

$$D = (\alpha_0 + \alpha_1\, |y(x)|)\,\frac{dy(x)}{dt} \tag{15}$$

The purpose of equation (15) is to explain the fact that reduction by one third of an excitation force on ship reduces by one half the vibration amplitude at resonance. A similar nonlinear representation of external damping has been proposed [114] for coupled horizontal-torsional vibration. Experimental evidence of the dependence of damping on amplitude is not yet adequate, however.

In finite element representations of a structure the damping matrix [C] is defined as a linear combination of mass and stiffness matrices. The normal mode approach is used:

$$[C] = \xi\,[M] + \zeta\,[K] \tag{16}$$

[M] and [K] are the mass and stiffness matrices. The logarithmic decrement is given by equation (17).

$$\delta_i = \frac{\xi\pi}{\omega_i} + \zeta\pi\omega_i \tag{17}$$

Correlations between calculations and measurements have shown that damping proportional to stiffness is the best representation of damping over a wide frequency range [38]. The constant of proportionality ζ is dependent to some extent upon the degree of idealization of the structure. For a three-dimensional idealization, ζ is close to 0.001; it approaches 0.0014 for a two-dimensional representation [13]. For a limited frequency range, damping is often taken as a constant fraction of critical damping C/C_c [28, 116]: the logarithmic decrement is shown in equation (18).

$$\delta_i = \frac{2\pi \frac{C}{C_c}}{\sqrt{1 - (\frac{C}{C_c})^2}} \tag{18}$$

Empirical Formulas

Several semi-empirical formulas for the logarithmic decrement have been based on full-scale measurements. The following equation [56] has been suggested for the two-node vertical vibrations of cargo ships and tankers of length L (from 80 to 200 meters)

$$\delta_2 = \frac{3.5}{L}$$

and for the higher modes (19)

$$\delta_i = \delta_2 \left(\frac{\omega_i}{\omega_2}\right)^{3/4}$$

Equation (19) is a modification of an earlier equation [125]. Equation (20) has been found to be a good approximation [45].

$$\delta_i = \omega_i \frac{60}{2\pi} [1.45 (y_S/y_T)_i + 2.15 (y_B/y_T)_i] \times 10^{-3} \quad (20)$$

In equation (20) y_S/y_T and y_B/y_T are ratios of shearing and bending deflections to total deflection for modes with up to eight nodes of a "representative" ship [45]. Hirowatari [31] proposed the formula

$$\delta_i = 1.065 \times 10^{-2} \omega_i^{1/2} \text{ for } \omega_i < 31.5 \text{ rad/sec} \quad (21)$$

Slamming test results have been used to obtain the relationship [2] shown in equation (22).

$$\delta_i = 7.3 \times 10^{-3} \omega_i \quad (22)$$

A comparison of the various equations used to calculate the logarithmic decrement is given in Figure 2. The data are for a 340,000 tdw tanker.

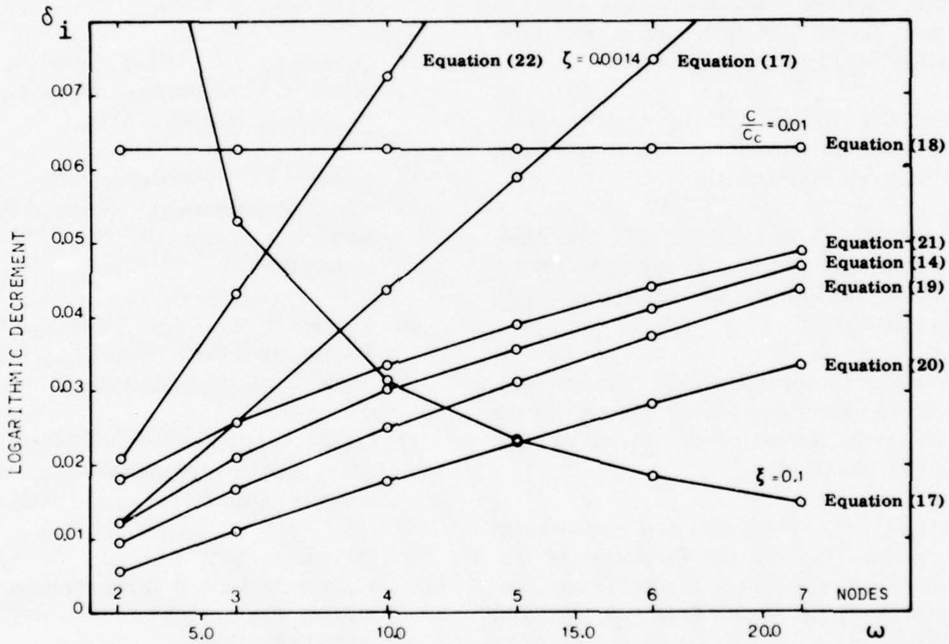

Figure 2. Logarithmic decrement versus cyclic frequency for a 340,000 tdw tanker.

REFERENCES

1. Aertssen, G. and deLembre, R., "Calculation and Measurement of the Vertical and Horizontal Vibration Frequency of a Large Ore Carrier," Trans. North East Coast Inst. Engr. Shipbldg., 86, pp 9-12 (1967-70).

2. Aertssen, G. and deLembre, R., "A Survey of Vibration Damping Factors Found from Slamming Experiments on Four Ships," Trans. North East Coast Inst. Engr. Shipbldg., 87, pp 83-86 (1970-71).

3. Anderson, G. and Norrand, K., "A Method for the Calculation of Vertical Vibration with Several Nodes and Some Other Aspects of Ship Vibration," Trans. RINA, 111, pp 367-383 (1969).

4. Bell, A.O. and Taylor, K.V., "Wave-Exctied Hull Vibration, Measurements on a 47,000 d.w.t. Tanker," Shipping World and Shipbuilder, 161, pp 412-419 (1968).

5. Bett, C.V., Bishop, R.E.D., and Price, W.G., "A Survey of Internal Hull Damping," Trans. RINA, preprint (w2) (1976).

6. Bishop, R.E.D. and Taylor, R.E.,"On Wave-Induced Stress in a Ship Executing Symmetric Motions," Phil. Trans. Roy. Soc. Lond., A275, pp 1-32 (1973).

7. Bourceau, G. and Volcy, C.G., "Forced Vibration Resonators and Forced Vibration of the Hull," Intl. Shipbldg. Prog., 18, pp 243-271 pp 275-294 (1971).

8. Breslin, J.P., "Theoretical and Experimental Techniques for Practical Estimation of Propeller-Induced Vibratory Forces," Trans. Soc. Naval Architects Marine Engr., 78, pp 23-40 (1970).

9. Breslin, J.P., "Exciting Force Operators for Ship Propellers," AIAA J. Aeronuat., 5 (3), pp 85-90 (1971).

10. Breslin, J.P., "Techniques for Estimating Vibratory Forces Generated by Propellers," Tech. and Res. Bull., Soc. Naval Architects Marine Engr. (Jan 1975).

11. Brönlund, O.E., "Eigenvalues of Large Symmetric Matrices," Proc. Fourth Intl. Ship Struc. Cong., Tokyo, pp 283-287 (1970).

12. Burrill, L.C., Robson, W., and Towsin, R.L., "Ship Vibration: Entrained Water Experiments," Trans. RINA, 104, pp 415-435 (1962).

13. Catley, D. and Norris, C., "Theoretical Prediction of the Vertical Dynamic Response of Ship Structures Using Finite Elements and Correlation with Ship Mobility Measurements," Proc. Eleventh Symp. Naval Hydrodynamics, Dept. Mech. Engr., Univ. College London, pp VI.23-VI.38 (Mar-Apr 1976).

14. Chowdhury, P.C., "Fluid Finite Elements for Added-Mass Calculations," Intl. Shipbldg. Prog. 19 (217), pp 302-309 (1972).

15. Collatz, L., "Eigenwertaufgaben mit technischen Anwendungen," Akademische Verlagsgesellschaft, Geest and Portig, 2. Auflage, Leipzig (1963).

16. Cowper, G.R., "The Shear Coefficient in Timoshenko's Beam Theory," J. Appl Mech., Trans. ASME, 33, p 335 (1966).

17. Csupor, D.A., "Zur Theorie un Berechnung der freien ungedämpften Schwingungen des Schiffskörpers," Diss. TH Hannover (1956).

18. Dawson, B. and Davies, M., "An Addition to Myklestad's Method Giving Convergence to a Natural Frequency," J. Ship Res., 19 (2), pp 130-132 (1975).

19. Frank, W., "Oscillating of Cylinders in or below the Free Surface of Deep Fluids," NSRDC, Washington, D.C., Rep. No. 2375 (1967).

20. Fried, I., "Condensation of Finite Element Eigenproblems," AIAA J., Tech. Note, 10 (11), pp 1529-1530 (1972).

21. Frivold, H., "Solid Boundary Factors for the Afterbody of an LNG Carrier," Norwegian Maritime Res., 4 (1), pp 16-20 (1976).

22. van Gent, W., "Unsteady Lifting-Surface Theory for Ship Screws: Derivation and Numerical Treatment of Integral Equation," J. Ship Res., 19 (4), pp 243-253 (1975).

23. Goodman, R.A., "Wave-Excited Main Hull Vibration in Large Tankers and Bulk Carriers," Trans. RINA, 113, pp 167-184 (1971).

24. Grim, A., "Hydrodynamic Masses," Schiffstechnik, 5 (29) (1958).

25. van Gunsteren, F.F., "Springing, Wave-Induced Ship Vibration," Intl. Shipbldg. Prog., 17 (195), pp 333-347.

26. Gupta, K.K., "Recent Advances in Numerical Analysis of Structural Eigenvalue Problems," in Theory and Practice in Finite Element Structural Analysis, Proc. 1973 Tokyo Seminar Finite Element Anal., pp 249-271.

27. Guyan, R.J., "Reduction of Stiffness and Mass Matrices," AIAA J., 3 (2), p 380 (1968).

28. Hansen, H.R. and Skaar, K.T., "Hull and Superstructure Vibrations, Design Calculation by Finite Elements," Proc. Symp. High Powered Propulsion Large Ships, Netherlands Ship Model Basin, Wageningen (1974).

29. Hart, H.H.'t, "Hull Vibrations of the Cargo-Liner Koudekerk," Intl. Shipbldg. Prog., 18 (206), pp 373-383 (1971).

30. Havelock, T.H., "Ship Vibrations: the Virtual Inertia of a Spheroid in Shallow Water," Quart. Trans. INA, 95 (1953).

31. Hirowatari, T., "Magnification Factors in the Higher Modes of Ship Vibration," JSNA-Japan, 113, pp 156-168 (1963).

32. Huse, E., "Hull Vibration and the Measurements of Propeller-Induced Pressure Fluctuations," Intl. Shipbldg. Prog., 17 (187), pp 87-95 (1970).

33. Huse, E., "Pressure Fluctuations on the Hull-Induced by Cavitating Propellers," Norwegian Ship Model Expt. Tank, Publ. No. 111 (Mar 1972).

34. Hylarides, S., "Lowest Natural Frequencies of Structures with Rigid-Body Degrees of Freedom," J. Ship Res., 12 (2), pp 131-136 (1968).

35. Hylarides, S., "Finite Element Technique in Ship Vibration Analysis," Intl. Shipbldg. Prog., 15 (169), pp 328-338 (1968).

36. Hylarides, S., "Recent Developments in Hull and Shaft Vibration Analysis," Intl. Shipbldg. Prog., 17 (190), pp 185-190 (1970).

37. Hylarides, S., "Hull Resonance: An Explanation of Excessive Vibrations," Intl. Shipbldg. Prog., 21 (236), pp 89-99 (1974).

38. Hylarides, S., "Damping in Propeller-Generated Ship Vibrations," Netherlands Ship Model Basin, No. 468 (1974).

39. Proc. First International Ship Structures Congress, Glasgow (1961).

40. Proc. Second International Ship Structures Congress, Delft, The Netherlands (1964).

41. Proc. Third International Ship Structures Congress, Oslo, Norway (1967).

42. Proc. Fourth International Ship Structures Congress, Tokyo, Japan (1970).

43. Proc. Fifth International Ship Structures Congress, Hamburg, West Germany (1973).

44. Johnson, A.J., "Vibration Tests of All-Welded and All-Riveted 10,000 Ton Dry Cargo Ship," Trans. North East Coast Inst. Engr., 67, pp 205-276 (1950-51).

45. Johnson, A.J., Ayling, P.W., and Couchman, A.J., "On the Vibration Amplitudes of Ships' Hulls," Trans. Instn. Engr. Shipbldg. Scotland, 105, pp 301-387 (1962).

46. Joosen, W.P.A. and Spangenberg, J.A., "On the Longitudinal Reduction Factor for the Added Mass of Vibrating Ships with Rectangular Cross-Section," Netherlands Research Centre, TNO, No. 40S (1961).

47. Kagawa, K. and Ohtaka, K., "Higher Mode Vertical Vibration of Giant Tanker," 2nd Rep., JSNA-Japan, 128, pp 295-309 (1970).

48. Kagawa, K. and Miyamoto, M., "Hull Vibration of Container Ship," Mitsubishi Tech. Bull. No. 81 (1972).

49. Kawakami, M. and Kiso, T., "On the Wave-Induced Ship Hull Vibration," JSNA-West Japan No. 51 (Mar 1976).

50. Kline, R.G. and Daidola, J.C., "Ship Vibration Prediction Methods and Evaluation of Influence of Hull-Stiffness Variation on Vibratory Response," Ship Struc. Committee, U.S. Coast Guard No. 249 (1975).

51. Kline, R.G. and Daidola, J.C., "Bibliography for Ship Vibration Prediction Methods and Evaluation of Influence of Hull-Stiffness Variation on Vibratory Response," Ship Struc. Committee, U.S. Coast Guard No. 250 (1975).

52. Kruppa, C., "Beitrag zum Problem der hydrodynamischen Trägheitsgrössen bei elastichen Schiffsschwingungen," Schiffstechnik, 9 (45), pp 38-60 (1962).

53. Kuiper, G., "Some Remarks on Lifting Surface Theory," Intl. Shipbldg. Prog., 18 (199), pp 131-148 (1971).

54. Kumai, T., "Shearing Vibrations of Ships," Eur. Shipbldg., 5 (2), pp 32-37 (1956).

55. Kumai, T., "The Effect of Loading Conditions on the Natural Frequency of Hull Vibration," Rep. R.I.A.M., Kyushu Univ. Japan, 5 (17), pp 9-20 (1957).

56. Kumai, T., "Damping Factors in the Higher Modes of Ship Vibration," Eur. Shipbldg., 7 (1), pp 29-34 (1958).

57. Kumai, T., "Added Mass Moment of Inertia Induced by Torsional Vibration of Ships," Rep. R.I.A.M., Kyushu Univ. Japan, 7 (28), pp 233-244 (1959).

58. Kumai, T., "Some Correction Factors for the Virtual Inertia Coefficient for the Horizontal Vibrations of a Ship," Rep. R.I.A.M., Kyushu Univ. Japan, 9 (33), pp 17-25 (1961).

59. Kumai, T. et al, "Measurement of Propeller Forces Exciting Hull Vibration by Use of Self-Propelled Model," Rep. R.I.A.M., Kyushu Univ. Japan, 9 (33), pp 1-15 (1961).

60. Kumai, T., "Some Aspect of the Propeller Bearing Forces Exciting Hull Vibration of a Single Screw Ship," Rep. R.I.A.M., Kyushu Univ. Japan, 9 (33), pp 27-34 (1961).

61. Kumai, T., "The Effect of Distribution of Load upon the Virtual Inertia Coefficient in the Vertical Vibration of a Ship," Rep. R.I.A.M., Kysuhu Univ. Japan, 10 (37) (1962).

62. Kumai, T., "On the Three-Dimensional Correction Factor for the Virtual Inertia Coefficient in the Vertical Vibration of Ships," JSNA-Japan, 112 (Dec 1962).

63. Kumai, T. and Ochi, Y., "On the Vibration of Ships in Wave," Rep. R.I.A.M., Kyushu Univ. Japan, 11 (40), pp 13-19 (1963).

64. Kumai, T., "On the Apparent Mass of Cargo Oil in Vibration of a Tanker," Rep. R.I.A.M., Kyushu Univ. Japan, 13 (46), pp 69-78 (1965).

65. Kumai, T., "Effect of Shear Deflection and Rotatory Inertia on the Damping of the Flexural Vibration of a Ship Hull," Rep. R.I.A.M., Kyushu Univ. Japan, 13 (46), pp 61-68 (1965).

66. Kumai, T. and Sakurada, Y., "On the Measurements of Propeller Surface Force of the Self-Propelled Model of a Tanker," Rep. R.I.A.M., Kyushu Univ. Japan, 13 (46), pp 79-95 (1965).

67. Kumai, T., "A Method for Evaluating the Three-Dimensional Reduction Factor of the Virtual Mass in the Vertical Vibration of Ships," Japan Shipbldg. Marine Engr., 1 (3), pp 15-21 (1966).

68. Kumai, T., "Vibration of a Mammoth Tanker with Special Consideration to Athwartship Flexibility," Eur. Shipbldg., 16 (3), pp 50-53 (1967).

69. Kumai, T., "On the Estimation of Natural Frequencies of Vertical Vibration of Ships," Rep. R.I.A.M., Kyushu Univ. Japan, 16 (54), pp 239-250 (1968).

70. Kumai, T. and Tasai, F., "On the Wave Exciting Force and Response of Whipping of Ships," Eur. Shipbldg., 19 (4), pp 42-47 (1970).

71. Kumai, T., "Wave-Induced Force Exciting Hull Vibration and Its Response," JSNA-West Japan, No. 44 (Aug 1972).

72. Kumai, T., "On the Three-Dimensional Entrained Water in Vibration of Lewis' Section Cylinder with Finite Length," JSNA-West Japan, No. 50 (Aug 1975).

73. Lamb, H., Hydrodynamics, Cambridge Univ. Press (1895).

74. Landweber, L. and Macagno, M., "Added Mass of Two-Dimensional Forms Oscillating in a Free Surface," J. Ship Res., 1 (3), pp 20-30 (1957).

75. Landweber, L. and Macagno, M., "Added Mass of a Three Parameter Family of Two-Dimensional Forms Oscillating in a Free Surface," J. Ship Res., 2 (4) (1959).

76. Landweber, L. and Macagno, M., "Added Mass of a Rigid Prolate Spheroid Oscillating Horizontally in a Free Surface," J. Ship Res., 3 (4), pp 30-36 (1960).

77. Landweber, L. and Macagno, M., "Added Mass of Two-Dimensional Forms by Conformal Mapping," J. Ship Res., 11 (2), pp 109-116 (1967).

78. Landweber, L., "Vibration of a Flexible Cylinder in a Fluid," J. Ship Res., 11 (3), pp 143-150 (1967).

79. Landweber, L., "Natural Frequencies of a Body of Revolution Vibrating Transversely in a Fluid," J. Ship Res., 15 (2), pp 97-114 (1971).

80. Lazan, B.J., Damping of Materials and Members in Structural Mechanics, Pergamon Press, London (1968).

81. Leibowitz, R.C. and Kennard, E.H., "Theory of Freely Vibrating Non-Uniform Beams, Including Methods of Solution and Application to Ships," David Taylor Model Basin Rep. 1317 (1961).

82. Lewis, F.M., "The Inertia of the Water Surrounding a Vibrating Ship," Trans., Soc. Naval Architects Marine Engr., 37, pp 1-20 (1929).

83. Lewis, F.M., "Propeller-Vibration Forces," Trans., Soc. Naval Architects Marine Engr., 71, pp 293-318 (1963).

84. Lewis, F.M., "Propeller Vibration Forces in Single-Screw Ships," Trans., Soc. Naval Architects Marine Engr., 77, pp 318-334 (1969).

85. Lin, Y.K., Probalistic Theory of Structural Dynamics, McGraw-Hill (1967).

86. Little, R.S. and Lewis, E.V., "A Study of Wave-Induced Bending Moments," Shipping World and Shipbuilder, 165 (3867), pp 357-360.

87. Madsen, N., "Vertikale Skrogsvingninger i Store Tankskibe," Dept. Ocean Engr., Tech. Univ. Denmark (1975).

88. Maeda, Y., "On the Coupled Vibration of the Longitudinal Vibration, Vertical Vibration and the Vibration of Shaft System," 2nd Rep., JSNA-West Japan, No. 44 (Aug 1972).

89. Mathewson, A.W., "Calculation of the Normal Vertical Flexural Modes of Hull Vibration by Digital Process," David Taylor Model Basin Rep. 706 (1950).

90. McGoldrick, R.T. and Russo, V.L., "Hull Vibration Investigation on SS Gopher Mariner," Trans., Soc. Naval Architects Marine Engr., 63, pp 436-475 (1955).

91. McGoldrick, R.T., "Ship Vibration," David Taylor Model Basin Rep. 1451 (1960).

92. Meirovitch, L., Analytical Methods in Vibrations, MacMillan, London (1967).

93. Mišra, Prayag Narayan, "Transverse Vibrations of a Ship Hull in Ideal Fluid, Determined through Variational Methods," J. Ship Res., 18 (3), pp 185-202 (1974).

94. Myklestad, N.O., "A New Method for Calculating Natural Modes of Uncoupled Bending Vibrations of Airplane Wings and Other Types of Beams," J. Aeron. Sci., 11, pp 153-162 (1944).

95. Nering, K. et al, "Theroetische und experimentelle Untersuchungen zur Beeinflussung der Resonanzschwingungen des Schiffskörpers ausgeführt an einer Serie von Kühl- and Transportschiffen," Schiffbauforschung, 13 (1/2), pp 1-13 (1974).

96. Ohtaka, K., Kumai, T., Ushijima, M., and Ohji, M., "On the Horizontal and Torsional Vibration of Ships," JSNA-Japan, 121 (June 1967).

97. Ohtaka, K., Kagawa, K., and Yamamoto, T., "Higher Mode Vertical Vibration of a Giant Tanker," JSNA-Japan, 125 (1969).

98. Ohtaka, K., "Vertical Vibration of Ships Coupled with Bottom Vibration," Mitsubishi Tech. Bull. No. 83 (1973).

99. Oossanen, P. van and Kooy, J. van der, "Vibratory Hull Forces Induced by Cavitating Propellers," Trans. RINA, 115, pp 111-144 (1973).

100. Oosterveld, M.V.C., Verdonk, C., Kooy, J. van der, and Oossanen, P. van der, "Some Propeller Cavitation and Excitation Considerations for Large Tankers," West European Conf. Mar. Tech. Proc., Delft (May 1974).

101. Palm, L., "Ermittlung der Schubsteifigkeit zur Berechnung höherer Eigenfrequenzen und Eigenformen von vertikalen Schiffskörperschwingungen," Schiffbauforschung, 14 (5/6), pp 153-162 (1975).

102. Pohl, K.H., "Die durch eine Schiffschraube auf benachbarten Platten erzeugten periodischen hydrodynamischen Drücke," Schifftechnik, 7 (35), pp 5-18 (1960).

103. Porter, W.R., "Pressure Distributions, Added-Mass and Damping Coefficients for Cylinders Oscillating in a Free Surface," Inst. Engr. Res., Univ. Cal. (1960).

104. Postl, R., "Elastische Schwingungen in der Schiffstechnik," Z. Agnew. Math. Mech., 52, pp T281-T287 (1972).

105. Prohaska, C.W., "Vibrations Verticales du Navire," Bull. de l'Association Technique Maritime et Aéronautique (1947).

106. Reed, F.E., "The Design of Ships to Avoid Propeller-Excited Vibrations," Trans., Soc. Naval Architects Marine Engr., 79, pp 244-280 (1971).

107. Restad, K., Volcy, G.C., Garnier, H., and Masson, J.C., "Investigations on Free and Forced Vibrations of an LNG Tanker with Overlapping Propeller Arrangement," Trans., Soc. Naval Architects Marine Engr., 81, pp 307-347 (1973).

108. Robinson, D.C., "Damping Characteristics of Ships in Vertical Flexure and Considerations in Hull Damping Investigations," David Taylor Model Basin, No. 1876 (Dec 1964).

109. Ruiz-Fornells, R. et al, "Hull Vibrations," Astilleros Espanoles, S.A. Symp. Paper, BSRA Trans. No. 3888 (1976).

110. Schade, H.A., "Effective Breadth of Stiffened Plating," Trans., Soc. Naval Architects Marine Engr., 59, pp 403-430 (1951).

111. Schade, H.A., "Effective Breadth Concept in Ship Structures," Trans., Soc. Naval Architects Marine Engr., 60, pp 410-430 (1953).

112. Schadlofsky, E., "The Calculation and Measurement of Elastic Natural Frequencies of Ship Hulls," Schiffbautechnische Gesellschaft, 13 (1932).

113. Schlick, O., "On the Vibration of Steam Vessels," Trans. INA (1884).

114. Schmitz, K-P, "Zum Problem der erzwungenen gekoppelten horizontal - torsionsschwingungen des Schiffskörpers," Schiffbauforschung, 13 (1/2), pp 14-19 (1974).

115. Sellers, M.L. and Kline, R.G., "Some Aspects of Ship Stiffness," Trans., Soc. Naval Architects Marine Engr., 75, pp 268-288 (1967).

116. Senjanović, I. and Skaar, K.T., "Problems of Coupling between Ship Hull and Substructure Vibration," HANSA, 112 (20), pp 1574-1578 (1975).

117. Senjanović, I. and Skaar, K.T., "Problems of Ship Vibration Present Solutions and Further Investigations," Trans., Soc. Naval Architects Marine Engr., Sp. Mtg. (June 1976).

118. Sezawa, K. and Watanabe, W., "Damping Forces in Vibration of a Ship," JSNA-Japan, 59 (1936).

119. Shioiri, J. and Tsakonas, S., Three-Dimensional Approach to the Gust Problem for a Screw-Propeller," J. Ship Res., 7 (4), pp 29-53 (1964).

120. Smith, C.S., "Buckling and Vibration of a Ships Vee- Bottom Structure," Trans. RINA, 116, pp 261-274 (1974).

121. Sparenberg, J.A., "Application of Lifting Surface Theory to Ship Screws," Koninkl. Ned. Akad. Wetenschap, Proc. Amsterdam, 62, Ser. B (1959).

122. Suetsugu, I. and Fujii, K., "The Effect of the Bottom Vibration on the Hull Natural Frequencies," Intl. Shipbldg. Prog., 10 (109), pp 351-364 (1963).

123. Tasai, F., "Formula for Calculating Hydrodynamic Force of a Cylinder Heaving on a Free Surface (n-Parameter Family)," Note Rep. RIAM, Kyushu Univ. Japan, 8 (31), pp 71-74 (1960).

124. Taylor, J.L., "Some Hydrodynamical Inertia Coefficients," Phil. Mag., 9 (1930).

125. Taylor, J.L., "Vibration of Ships," Trans. INA (1930).

126. Todd, F.H., "Some Measurements of Ship Vibration," Trans. North East Coast Instn. Engr. Shipbldg., 48, pp 65-88 (1931-32).

127. Todd, F.H., Ship Hull Vibration, Edw. Arnold, London (1961).

128. Townsin, R.L., "Virtual Mass Reduction Factors, J'Values for Ship Vibration Calculations Derived from Tests with Beams Including Ellipsoids and Ship Models," Trans. RINA, 111, pp 385-397 (1969).

129. Tsakonas, S., Breslin, J.P., and Miller, M., "Correlation and Application of an Unsteady Flow Theory for Propeller Forces," Trans., Soc. Naval Architects Marine Engr., 75, pp 158-180 (1967).

130. Tsakonas, S., Jacobs, W.R., and Rank, P.H., "Unsteady Propeller Lifting-Surface Theory with Finite Number of Chordwise Modes," J. Ship Res., 12 (1), pp 14-45 (1968).

131. Tsakonas, S. and Jacobs, W.R., "Propeller Loading Distributions," J. Ship Res., 13 (4), pp 237-257 (1969).

132. Tsakonas, S. et al, "Documentation of a Computer Program for the Pressure Distribution, Forces and Moments on Ship Propellers in Hull Waves," Stevens Inst. Tech., Rep. SIT-DL-76-1863 (1976).

133. Verbrugh, P.J., "Unsteady Lifting Surface Theory for Ship Screws," Netherlands Ship Model Basin Rep. 68-036-AH, 62 (5) (1968).

134. Vorus, W.S., "A Method for Analyzing the Propeller-Induced Vibratory Forces Acting on the Surface of a Ship Stern," Trans., Soc. Naval Architects Marine Engr., 82, pp 186-198 (1974).

135. Ward, G. and Willshare, G.T., "Propeller-Excited Vibration with Particular Reference to Full-Scale Measurements," RINA Sp. Mtg., Paper No. 4 (1975).

136. Wereldsma, R., "Tendencies of Marine Propeller Shaft Excitations," Intl. Shipbldg. Prog., 19 (218), pp 328-332 (1972).

137. Wereldsma, R. and Moeyes, G., "Wave and Structural Load Experiments for Elastic Ships," Eleventh Symposium Naval Hydrodynamics, Dept. Mech. Engr., Univ. College London, pp VI.3-VI.15 (Mar-Apr 1976).

138. Yamakoshi, M. and Ohnuma, S., "On the Coupling of Hull Vibration and Bottom Vibration of Ships," JSNA-Japan, 118 (Dec 1965).

INA: Institution of Naval Architects, London

RINA: Royal Institution of Naval Architects

JSNA: Journal of the Society of Naval Architects, Tokyo

RIAM: Research Institute for Applied Mechanics

BSRA: British Ship Research Associates

BOOK REVIEWS

SECRETS OF NOISE CONTROL

A. Thumann and R.K. Miller

Fairmont Press, Atlanta, Georgia

The catchy title raises some unwarranted expectations. The book does not reveal any secrets, nor should one expect to have deep secrets revealed so easily. The best way I can characterize the book is by calling it a "cookbook for people with simple tastes." I would hasten to add that it is unreasonable to expect all people with noise problems to desire to become "gourmets" in noise control engineering. There are undoubtedly many people who hunger for information on noise control engineering and whose tastes are, of necessity, quite simple.

The book is aimed at providing simple recipes specifically for people who need basic information on noise control for the purpose of planning manufacturing facilities. That the book does not go beyond this limited purpose may be an advantage for readers with this need. The information in the book was gathered from many sources and can be understood by the nonspecialist. Sources of material taken from the literature are listed at the end of the book. I consider the reference list inadequate, however, because it provides little guidance for those who might want additional information. Unfortunately, the text implies that the book contains all there is to know about noise control, and this may give the reader a false sense of security.

This second edition contains very few changes or corrections. Unfortunately, the errors that were pointed out in the review of the first edition (Noise Control Engineering, Jan-Feb 1975, p 48) have not been corrected. On page 198, for instance, the typical driving force frequency for a transformer is still given as 60 Hz, although on page 93 it is stated correctly that "most of the noise of a transformer is radiated in the 120, 240, and 360 Hz harmonics...."

The main change I discovered is that the new edition estimates the consulting fees for an acoustic consultant on a job requiring 702 hours of effort to be $21,060; estimate of the fee for the same job in the first edition was $10,530. Whether the hourly rate is $15 or $30, the reader will no doubt realize that an acoustical consulting job of such magnitude requires more insight into the "secrets of noise control" than can be learned from any book.

Dr. Peter K. Baade
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ELEMENTS OF VIBRATION ANALYSIS

L. Meirovitch

McGraw-Hill Book Company, New York, 1975

Many books on vibration analysis have appeared in the past 15 years. This is one of the better ones. It is an updated version of Analytical Methods in Vibrations. Some information has been deleted, and some has been updated. Elementary discussions of finite element theory and nonlinear vibrations have been added; computer applications are emphasized.

Chapters I and II consider single-degree-of-freedom systems and response due to external stimuli, including periodic, harmonic, and nonperiodic. Examples and applications of standard and integral transform methods used in response analysis are given.

Chapter III considers two-degree-of-freedom systems. Examples include coordinate transformation-coupling, beat phenomena, response to harmonic excitation, and analysis of undamped vibration absorbers.

The heart of vibration analysis is multi-degree-of-freedom systems. Chapter IV stresses matrix applications, linear transformation, eigenvalue problems, and orthogonality of modal vectors. Modal analysis is applied to problems involving the response of systems to initial excitation, influence coefficients, and eigenvalue solutions with the power method. The Rayleigh method is considered. The reviewer was disappointed that little mention was made of the Jacobi method; no mention was made of other useful methods of eigenvalue solutions.

Chapter V describes continuous systems; beam column effects are neglected. The author shows the versatility of modal analysis in continuous beams and rods. The reviewer would have liked a discussion of the application of impedance methods applied to large structures.

Chapters VI and VII introduce the reader to analytical dynamics. The Lagrangian method is described, but no mention is made of Hamilton's equations. Approximate methods are used to solve problems involving continuous systems. The Rayleigh-Ritz method, the Rayleigh method, the Holzer method for torsional vibration analysis, and the lumped parameter method are considered. The Myklestad-Prohl method, which is presently being used in a number of industrial vibration analyses programs, is not included, however.

In Chapter VIII the author acknowledges that the finite element method has become an important part of vibration analysis. This method has been neglected in a number of books on vibration analysis. Stiffness methods for beams and columns are applied to system response in a short but adequate section. The reviewer would have preferred to see descriptions of the Houbolt and the Wilson methods.

Chapters IX and X provide a good introduction to nonlinear systems. The following methods are described: the phase-plane method, Routh-Hurwitz criteria (only real equations), perturbation methods (Van der Pol's equation), Lindstet's method, Krylov-Bogolibov-Mitropolsky method (harmonic balance), and Mathieu's equation. In concludes with a discussion on sub-harmonics.

Chapter XI is about random vibration theory and contains an excellent discussion on amplitude probability distribution (Gaussian and Rayleigh), auto- and cross spectral-densities, auto- and cross-correlations, and coherence. Response of simple and complex systems is included. The appendices include Fourier series, Laplace transforms, and the matrix method.

The reviewer was impressed with the book and considers it an excellent text on vibration theory. Both students and experienced engineers will find the book useful. The major criticism of the book is that, although computer applications are emphasized, no simple computer programs are included to illustrate the theory.

Herb Saunders
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SHORT COURSES

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STRATEGY OF EXPERIMENTATION WORKSHOP/SEMINAR

Dates: June 1977

Place: Denver/Detroit/Minneapolis

Objective: This course is designed to develop skills and confidence in using several experimental strategies in realistic problem situations. Practice in applying every experimental design taught in the course helps solidify the learning experience so that you can use the skills immediately. Applicants should have a college degree or extensive experience in their field. The dates for the different locations are:

- Denver, CO — June 7-9
- Detroit, MI — June 14-16
- Minneapolis, MN — June 21-23

Contact: J. R. Coulson, Manager, Seminar Services, E. I. du Pont de Nemours & Co., F&F Dept., Applied Technology Div., Brandywine Bldg., Wilmington, DE 19898 Tele. (302) 774-6406

JULY

THIRD INTERNATIONAL OCEAN ENGINEERING AND MANAGEMENT COURSE

Dates: July 11-22, 1977

Place: UCLA Campus, Los Angeles, California

Objective: To provide an annual educational forum for technology transfer between engineering and management working in the field of ocean technology development.

Contact: Ocean Engineering and Management Course, 6266 Boelter Hall, UCLA Extension, Los Angeles, CA 90024 Tele. (213) 825-3858

DYNAMIC ANALYSIS OF OFFSHORE STRUCTURES

Dates: July 18-22, 1977

Place: UCLA Campus, Los Angeles, California

Objective: This course will stress practical techniques of dynamic (random data) analysis and structural measurements for determining dynamic properties of offshore structures, such as platforms, caissons, risers, etc. These methods are based upon applications of spectral density, frequency response, coherence functions and other statistical quantities which can be computed from available measured data.

Contact: Ocean Engineering and Management Course, 6266 Boelter Hall, UCLA Extension, Los Angeles, CA 90024 Tele. (213) 825-3858

INSTRUMENTATION FOR MECHANICAL ANALYSIS

Dates: July 25-29, 1977

Place: University of Michigan, Ann Arbor, MI

Objective: Emphasis is on the use of instruments by non-electrical engineers to analyze systems. Attendees will use a wide range of transducers and associated instrumentation. Morning lectures are devoted to theory and afternoons to various applications in the laboratory. Previous instrumentation experience is not required.

Contact: Engineering Summer Conferences, 200 Chrysler Center, North Campus, The University of Michigan, Ann Arbor, MI 48109

AUGUST

NOISE CONTROL IN ENGINEERING

Dates: August 8 - 12, 1977
Place: University of Michigan, Ann Arbor, MI
Objective: Risk of hearing damage from factory noise (e.g., OSHA regulations) and excessive product noise (e.g., EPA regulations) constitute serious concerns for industry. This course provides engineers and managers with comprehensive knowledge of noise-control engineering and criteria for application to practical problems.

Contact: Engineering Summer Conferences, 200 Chrysler Center, North Campus, The University of Michigan, Ann Arbor, MI 48109

STATIC AND DYNAMIC FINITE ELEMENT ANALYSIS WITH COMPUTER WORKSHOP

Dates: August 8 - 12, 1977
Place: Massachusetts Institute of Technology Cambridge, Massachusetts
Objective: The objective in this program is to present the essential details of selecting an appropriate finite element model, analyzing the model, and interpreting the results. This is achieved by a coordinated set of lectures and a computer workshop.

Contact: Office of the Summer Session, Room E19-356, Massachusetts Institute of Technology, Cambridge, MA 02139 Tele. (617) 253-2101

IMPEDANCE AND DYNAMIC ANALYSIS OF STRUCTURES

Dates: August 15-19, 1977
Place: State College, PA
Objective: To present measurement and analysis techniques by which impedance, transfer, and impulse functions are used directly in prediction of structural response to ground motion or pressure loading, in fluid-structure interaction, in identifying modal properties of structures, and in arriving at finite element models.

Contact: Dr. V.H. Neubert, 133 Hammond Bldg., The Pennsylvania State University, University Park, PA 16802 Tele. (814) 6161

THE SCIENTIFIC AND MATHEMATICAL FOUNDATIONS OF ENGINEERING ACOUSTICS

Dates: August 15 - 26, 1977
Place: Massachusetts Institute of Technology Cambridge, Massachusetts
Objective: This program is a specially developed course of study which is based on two regular MIT subjects (one graduate level and one undergraduate level) on vibration and sound in the Mechanical Engineering Department. The program emphasizes those part of acoustics -- the vibration of resonators, properties of waves in structures and air -- the generation of sound and its propagation that are important in a variety of fields of application. The mathematical procedures that have been found useful in the processing of data are also studied.

Contact: Richard H. Lyon, Massachusetts Institute of Technology, Rm. 3-366, Dept. of Mech. Engrg., Cambridge, MA 02139

CORRELATION AND COHERENCE ANALYSIS FOR ACOUSTICS AND VIBRATION PROBLEMS

Dates: August 29 - September 2, 1977
Place: UCLA, Los Angeles, California
Objective: This course covers the latest practical techniques of correlation and coherence analysis -- ordinary, multiple, and partial -- for solving acoustics and vibration problems in physical systems.

Contact: Continuing Education in Engineering and Mathematics, Short Courses, 6266 Boelter Hall, UCLA Extension, Los Angeles, CA 90024 Tele. (213) 825-1047

FINITE ELEMENT ANALYSIS WORKSHOP

Dates: September 29 and 30, 1977
Place: Chicago, IL
Objective: This course covers the finite element modeling of machines and structures. The theory and practice of the finite element method will be discussed along with pertinent examples and case histories. Participants will be able to work their own problems.

Contact: Dr. Ronald L. Eshleman, Vibration Institute, Suite 206, 101 W. 55th St., Clarendon Hills, IL 60514 Tele. (312) 654-2254/654-2053

ABSTRACTS FROM THE CURRENT LITERATURE

Copies of articles abstracted in the DIGEST are not available from the SVIC or the Vibration Institute (except those generated by either organization). Inquiries should be directed to library resources. Government reports can be obtained from the National Technical Information Service, Springfield, VA 22151, by citing the AD-, PB-, or N- number. Doctoral dissertations are available from University Microfilms (UM), 313 N. Fir St., Ann Arbor, MI; U. S. Patents from the Commissioner of Patents, Washington, D.C. 20231. Addresses following the authors' names in the citation refer only to the first author. The list of periodicals scanned by this journal is printed in issues 1, 6, and 12.

ABSTRACT CONTENTS

ANALYSIS AND DESIGN

ANALYTICAL METHODS

(Also see No. 1017)

77-851

An Approximate Analysis of Transient Response of Time-Dependent Linear Systems by Use of Orthogonal Polynomials

S.C. Sinha and C.C. Chou

Biomechanics Research Ctr., College of Engrg., Wayne State Univ., Detroit, MI 48202, J. Sound Vib., 49 (3), pp 309-326 (1976) 34 figs, 3 refs

Key Words: Transient response, Linear systems

The paper deals with the approximate analysis of second order linear systems with variable coefficients through the application of orthogonal polynomials. The time-dependent functions appearing as coefficients in the system equations may be periodic, non-periodic or can have multiple turning points.

77-852

Formulations and Solution Procedures for Nonlinear Structural Analysis

J.A. Stricklin and W.E. Haisler

Aerospace Engrg. Dept., Texas A&M Univ., College Station, TX 77843, Intl. J. Computers and Structures, 7 (1), pp 125-136 (Feb 1977) 3 figs, 66 refs

Key Words: Dynamic structural analysis, Computer programs

This paper presents a survey of the formulations and solution procedures for nonlinear static and dynamic structural analysis. The formulations covered include the pseudo force method, the total Lagrangian method, the updated Lagrangian method, and the convected coordinate method. The relationship of each principle to the basic principle of virtual work is presented. For static analysis, the solution by direct minimization of the total potential, Newton-Raphson and modified Newton-Raphson, and the first and second order self correcting method are reviewed and put in proper perspective. For dynamic nonlinear analysis, a new method based on modal analysis using the pseudo force method is presented. Numerical results for the highly nonlinear dynamic response of a shallow cap under a step load at the apex shows the method to be 5 times faster than the Houbolt solution procedure. Other methods surveyed include the Newmark β method, the Wilson method, central differences, and the stiffly stable solution procedure of Park.

77-853

A Review of Substructure Coupling Methods for Dynamic Analysis

R.R. Craig and C.J. Chang

Texas Univ., Austin, TX, In: NASA, Langley Res. Center, Advan. in Eng. Sci., Vol. 2, 1976, pp 393-408 (see N77-10265 01-31)

N77-10267

Key Words: Substructure technique, Reviews

The state of the art is assessed in substructure coupling for dynamic analysis. A general formulation, which permits all previously described methods to be characterized by a few constituent matrices, is developed. Limited results comparing the accuracy of various methods are presented.

77-854

Bond Graphs for Nonholonomic Dynamic Systems

F.T. Brown

Dept. of Mech. Engrg., Lehigh Univ., Bethlehem, PA 18015, J. Dyn. Syst., Meas. and Control., Trans. ASME, 98 (4), pp 361-366 (Dec 1976) 9 figs, 6 refs

Key Words: Bond graphs, Holonomic systems, Nonholonomic systems

Two very different dynamic systems, one holonomic and the other nonholonomic, can have identical expressions for generalized kinetic energy, generalized potential energy, and transformational constraints between the generalized velocities, and therefore might be confused. Bond graphs for a broad class of nonholonomic systems are shown to differ from their holonomic counterparts simply by the deletion of certain gyrators. Simple examples suggest the engineering significance of nonholonomic systems.

OPTIMIZATION TECHNIQUES

77-855

Structural Optimization with Flutter Speed Constraints Using Maximized Step Size

R.F. O'Connell, N.A. Radovcich and H.J. Hassig

Lockheed-California Co., Burbank, CA 91500, J. Aircraft, 14 (1), pp 85-89 (Jan 1977) 2 figs, 9 refs

Key Words: Flutter, Optimization

A procedure is presented for the minimization of structural mass while satisfying flutter speed constraints. The procedure differs from other optimization methods in that the flutter speed is exactly satisfied at each resizing step, and the step size is determined by a direct minimization of the objective function (mass) for each set of flutter derivatives calculated. In conjunction with this method, a new move vector is suggested which results in an efficient resizing procedure.

FINITE ELEMENT MODELING

(Also see No. 884)

77-856

Efficient Large Scale Non-Linear Transient Analysis by Finite Elements

T. Belytschko, R.L. Chiapetta and H.D. Bartel
Dept. of Materials Engrg., Univ. of Illinois, Chicago, IL., Intl. J. Numer. Methods Engr., 10 (3), pp 579-596 (1976) 11 figs, 19 refs

Key Words: Transient response, Finite element technique, Interaction: structure-medium, Seismic design

An efficient computational scheme has been developed for transient, nonlinear analysis of structure-media interaction problems by finite elements. This 'direct' method is both faster and requires less core than fictitious force or tangential stiffness methods in explicit temporal integration schemes, hence it permits the solution of bigger and more complex problems. Techniques have been presented for including sliding-debonding interfaces and artificial viscosity, which is effective in reducing the spurious oscillations observed in explicitly integrated discrete systems.

MODELING

77-857

Modeling of Vibrating Systems - An Overview. Part II. Approximate Methods, Mechanical Impedance and Mobility, Transfer-Matrix Method, Finite Element Methods and Bond Graphs

A.J. Hannibal
Advanced Systems Dept., Lord Kinematics, Erie, PA 16512, Shock Vib. Dig., 8 (12), pp 11-20 (Dec 1976) 7 figs, 64 refs

Key Words: Mathematical models, Mechanical impedance, Transfer matrix methods, Finite element technique, Bond graphs, Reviews

A number of prominent modeling methods are presented in capsule form. The "type" of modeling technique is emphasized rather than details of the method. The references have been divided into sections relating to the modeling methods.

DIGITAL SIMULATION

(See No. 890)

PARAMETER IDENTIFICATION

77-858

A Recursive On-Line Estimation Method with Application to Aircraft Dynamics Parameter Identification

M. Sidar
Armament Development Authority, Israel Ministry of Defense, Israel J. Tech., 14 (1-2), pp 56-65 (1976) 6 figs, 16 refs

Sponsored by NRC and NASA

Key Words: Aircraft, Parameter identification

The problem of identifying constant system parameters and identifying and tracking variable parameters in multi-input, multi-output, linear and nonlinear systems is considered in this paper. An identification algorithm is developed on the basis of the one step prediction error concept using the minimum variance and the maximum likelihood approach. The identification is performed by applying the "output error" approach. The novel iterative algorithm, leading to recursive identification and tracking of the unknown parameters and the noise covariance matrix, is developed and presented here. Agile tracking, accurate, consistent and unbiased parameter estimates are obtained. Necessary conditions for a stable identification process are provided. Among different cases studied, special emphasis was focused on the aircraft dynamics identification problem, the stability and control derivatives of aircraft being identified. Some results are shown as examples in this paper.

77-859

Vibratory Identification of Beam Boundary Conditions

A.L. Sweet, J. Genin and P.F. Mlakar
Purdue Univ., W. Lafayette, IN, J. Dyn. Syst. Meas. and Control, Trans. ASME, 98 (4), pp 387-394 (Dec 1976) 5 figs, 5 refs

Key Words: Beams, System identification, Vibration measurement, Nondestructive testing, Buckling

In previous work, by means of stochastic and deterministic models, the authors presented a system identification theory for performing nondestructive testing of elastic systems. The procedure requires identification of the structure's boundary conditions. Herein, the mathematical models are improved, and vibration data are presented for the determination of the boundary parameters. These experimentally derived results are shown to validate the models.

DESIGN TECHNIQUES

(See No. 1035)

CRITERIA, STANDARDS, AND SPECIFICATIONS

77-860

Stationary Test Method for Noise From a Single Road Vehicle

C.G. Balachandran, K.O. Ballagh and T.A. Lister
Physics and Engrg. Lab., Lower Hutt, New Zealand,
Appl. Acoust., 10 (1), pp 49-56 (Jan 1977) 10 figs, 7 refs

Key Words: Traffic noise, Testing techniques, Regulations

Legislation in several countries restricting noise emitted by individual road vehicles recommends the use of international standard ISO R362, or something very similar, as a standard. This is a pass-by test with strict conditions laid down regarding the test site and weather conditions. As such it is not well suited for simple routine and quick roadside enforcement of vehicle noise. This paper describes some initial work aimed at clarifying and resolving a number of the difficulties commonly encountered with stationary tests and describes a procedure which gives good correlation with the ISO pass-by test and may lead to a test which could be used for routine service check and roadside enforcement.

77-861

On Using the ISO Standard to Evaluate the Ride Quality of Broad-Band Vibration Spectra in Transportation Vehicles

C.C. Smith
Dept. of Mech. Engrg., Univ. of Texas at Austin, Austin, TX,
J. Dyn. Syst. Meas. and Control, Trans. ASME, 98 (4), pp 440-443 (Dec 1976) 4 figs, 5 refs

Key Words: Vibration measurement, Ground effect machines, Automobile seats, Floors, Standards

The International Standards Organization "Guide for the Evaluation of Human Exposure to Whole-Body Vibrations," ISO 2631, is converted to a form usable for direct comparison with vibration data represented in power spectral density form. Comparisons are made between the ISO standard, the Urban Tracked Air Cushion Vehicle (UTACV) specification, and measured vibrations at the floorboard and seat of an automobile over smooth and rough roads. The data indicate that the ISO standard is less restrictive than the UTACV specification, and generally not restrictive enough to indicate the roughness of an automobile ride on a rough country road.

SURVEYS

77-862

Laboratory Measurements in Acoustics

C.R. Voorhees and D.S. Pallett
Applied Acoustics Section, Inst. for Basic Standards, National Bureau of Standards, Washington, D.C.,
Noise Control Engr., 7 (2), pp 52-56 (Sept/Oct 1976) 7 figs, 10 refs

Key Words: Test facilities, Reviews, Noise measurement

Results of a survey of 120 acoustical laboratory facilities are presented and discussed. The survey covered use of facilities for sound power, sound absorption, transmission loss and impact noise measurement, and the use of reference sound sources and anechoic chambers.

TUTORIAL

77-863

Guidelines for the Planning and Preparation of Illustrated Technical Talks

H.H. Hubbard
NASA, Langley Research Center, Hampton, VA 23665,
J. Acoust. Soc. Amer., 60 (5), pp 995-998 (Nov 1976) 2 figs, 3 refs

Key Words: Surveys

Guidelines are presented for the preparation of illustrated talks which are audience oriented and which are aimed at the efficient transfer of technical information. Early decisions concerning the required number of slides are helpful in initial planning for a good-quality talk. Detailed considerations are the establishment of limited objectives, selection of appropriate slide material, development of a text which is well coordinated with the slides, and accurate timing.

MODE SYNTHESIS AND ANALYSIS

(Also see Nos. 857, 904, 954, 960)

77-864

The Use of Modal Superposition in Nonlinear Dynamics

N.F. Morris

Dept. of Civil Engrg., Manhattan College, Riverdale, NY 10471, Intl. J. Computers & Structures, 7 (1), pp 65-72 (Feb 1977) 6 figs, 4 refs

Key Words: Modal superposition method, Nonlinear response, Cables (ropes), Suspension bridges, Computer programs

This paper describes the application of the modal superposition method to the calculation of the nonlinear dynamic response of structures. Although this method has often been used in the analysis of linear response, it was seldom applied in nonlinear problems. The reasons for this are obvious and seem to be quite valid, but the examples considered herein indicate that modal superposition may be a more useful computational tool than has heretofore been imagined. Four different problems are considered in this paper: a three dimensional cable stayed bridge, a double layered three dimensional cable network, an unstiffened suspension bridge, and a three dimensional elastic-plastic frame. The problems are chosen to exemplify both the strengths and weaknesses of the modal superposition method.

77-865

An Extension of the Southwell-Dunkerley Methods for Synthesizing Frequencies. Part I: Principles

M. Endo and O. Taniguchi

Dept. of Physical Engrg., Tokyo Inst. of Technology, Ookayama, Meguro-ku, Tokyo, Japan, J. Sound Vib., 49 (4), pp 501-516 (Dec 22, 1976) 6 figs, 7 refs

Key Words: Frequency synthesis, Single-degree-of-freedom systems

A new method for synthesis of isolated frequencies of composite vibrating systems is presented. First, from an analogy with single-degree-of-freedom systems composed of springs and masses, the essential patterns of synthesis are clearly recognized in regard to the composite, continuous systems to which the well-known Southwell-Dunkerley methods are applicable. Then, in a converse manner, another fundamental pattern of synthesis is formulated for a single-degree-of-freedom model with springs in series, and this leads to a general formulation of the new synthetic method for continuous systems.

77-866

An Extension of the Southwell-Dunkerley Methods for Synthesizing Frequencies. Part II: Applications

M. Endo and O. Taniguchi

Dept. of Physical Engrg., Tokyo Inst. of Technology, Ookayama, Meguro-ku, Tokyo, Japan, J. Sound Vib., 49 (4), pp 517-533 (Dec 22, 1976) 13 figs, 5 refs

Key Words: Frequency synthesis, Beams, Bars, Plates

The series-type synthetic method of obtaining frequencies and the combined synthetic one are examined mainly from the practical point of view, for some typical vibrating models. The fundamental frequency of a two-degree-of-freedom system with two springs and two masses, connected alternately in series is discussed. The accuracy of the approximate solutions is investigated without the influence of the error associated with Rayleigh's principle. The convergence of the approximate solutions as finite power-series expansions in the perturbation parameter is also studied. For two kinds of fairly complicated models composed of a beam, springs and concentrated masses as examples, a study is presented of the degree of the error of approximation accumulated when synthesizing many isolated frequencies on the basis of the series-type synthetic method or the combined synthetic one.

COMPUTER PROGRAMS

GENERAL

77-867

Frequency Domain Computer Programs for Prediction and Analysis of Rail Vehicle Dynamics. Volume 1. Technical Report

A.B. Perlman and F.P. DiMasi

Transportation Systems Center, Cambridge, MA, Rept. No. DOT-TSC-FRA-75-16.1, FRA/ORD-76/135.1, 116 pp (Dec 1975)

PB-259 287/1GA

Key Words: Computer programs, Railroad trains

Frequency domain computer programs developed or acquired by TSC for the analysis of rail vehicle dynamics are described in two volumes. Volume I defines the general analytical capabilities required for computer programs applicable to single rail vehicles and presents a detailed description of frequency domain programs developed at TSC in terms of analytical capabilities, model description, equations of motion, solution procedure, input/output parameters, and program control logic. Descriptions of programs FULL, FLEX, LATERAL, and HALF are presented. TSC programs for assessing lateral dynamic stability of single rail vehicles are also described.

77-868

Frequency Domain Computer Programs for Prediction and Analysis of Rail Vehicle Dynamics. Volume II: Appendixes

A.B. Perlman and F.P. DiMasi

Transportation Systems Center, Cambridge, MA, Rept. No. DOT-TSC-FRA-75-16.11, FRA/ORD-76/135.11, 102 pp (Dec 1975)

PB-259 288/9GA

Key Words: Computer programs, Railroad trains

Frequency domain computer programs developed or acquired by TSC for the analysis of rail vehicle dynamics are described in two volumes. Volume 2 contains program listings including subroutines for the four TSC frequency domain programs described in Volume 1.

77-869

Aeroelastic Analysis for Helicopter Rotor Blades with Time-Variable, Non-Linear Structural Twist and Multiple Structural Redundancy; Mathematical Derivation and Program User's Manual

R.L. Bielawa

United Technologies Research Center, East Hartford, CT., Rept. No. NASA-CR-2638, 155 pp (Oct 1976)

N77-10556

Key Words: Computer programs, Rotary wings, Helicopters, Blades

The differential equations of motion for the lateral and torsional deformations of a nonlinearly twisted rotor blade in steady flight conditions together with those additional aeroelastic features germane to composite bearingless rotors are derived. The differential equations are formulated in terms of uncoupled (zero pitch and twist) vibratory modes with exact coupling effects due to finite, time variable blade pitch and, to second order, twist. Also presented are derivations of the fully coupled inertia and aerodynamic load distributions, automatic pitch change coupling effects, structural redundancy characteristics of the composite bearingless rotor flexbeam-torque tube system in bending and torsion, and a description of the linearized equations appropriate for eigensolution analyses. Three appendixes are included presenting material appropriate to the digital computer program implementation of the analysis, program G400.

77-870

Signals Analysis Program 4 (SAPG 4): Computer Software Developed for the Extraction of Various Signal Parameters from Fatigue Inducing Loading Spectra

J.T.D. Fritz

Strength Mechanics Div., Council for Scientific and Industrial Research, Pretoria, South Africa, Rept. No. CSIR-ME-1408, 105 pp (Sept 1975)

N77-10554

Key Words: Computer programs, Random excitation

Computer software was developed to extract various desriptive and counting parameters from any random loading spectrum. The theory and algorithms used in determining various spectrum parameters are described and information on the various input and output data formats is given.

ENVIRONMENTS

ACOUSTIC

(Also see Nos. 860, 870, 905, 917, 933, 948
970, 971, 994, 999, 1010, 1011)

77-871

Resonance Absorbers with Additional Damping

E. Hirschwehr

Institut fuer Niederfrequenztechnik, Tech. Univ. Wien, Vienna, Austria, Acustica, 36 (4), pp 294-300 (Dec 1976) 5 figs, 7 refs
(In German)

Key Words: Acoustic absorption, Porous materials

Simplified electrical equivalent circuit diagrams are developed from impedance measurements on porous layers in a measurement tube, and give a rough approximation to the impedance behavior of porous layers. The effect of these porous layers on the resonance absorber is calculated and compared with the measured results.

77-872

Application of the Principle of Huygens to Active Sound Absorbers. I. Theory of Active Absorbers

G. Mangiante

Laboratoire de Mecanique et d'Acoustique, Equipe A3, 13274 Marseille Cedex 2, Acustica, 36 (4), pp 287-293 (Dec 1976) 6 figs, 10 refs
(In French)

Key Words: Acoustic absorption, Active absorption

This paper proposes, for three-dimensional sound propagation, a general theory of active sound absorption using Huygens principle. An attenuation coefficient for pure tones and complex sounds is studied, which illustrates the effects of errors on the attainable reduction of noise in an incident sound.

77-873

Multiple Scattering Between a Cylinder and a Plane

J.C. Bertrand and J.W. Young

U.S. Naval Undersea Ctr., San Diego, CA 92132, J. Acoust. Soc. Amer., 60 (6), pp 1265-1269 (Dec 1976) 9 figs, 3 refs

Key Words: Acoustic scattering

Experimental measurements of the backscattering of a plane acoustic wave by a rigid cylinder parallel to an elastic plate are compared with theoretical calculations. Two sets of calculations, one which includes the effects of multiple scattering between the cylinder and the plate and one which ignores these effects, were performed. Comparisons between the theories and experiment were made both for the case of the cylinder in front of the plate and for the case of the cylinder behind the plate.

77-874

On the Radiation of Sound from Baffled Finite Panels

P. Leehey

Acoustics and Vibration Lab., Massachusetts Inst. of Tech., Cambridge, MA, In: NASA, Langley Res. Ctr. Advan. in Eng. Sci., Vol. 3, 1976, pp 1043-1055 (see N77-10305 01-31)
N77-10325

Key Words: Baffles, Plates, Panels, Elastic waves, Sound waves

Theoretical and experimental research on structural-acoustic interaction is reviewed. The emphasis is upon the radiation from and acoustic loading of baffled rectangular plates and membranes. The topics discussed include a criterion for strong radiation loading, the mass law for a finite panel, numerical calculation of the radiation impedance of a finite panel in the presence of a parallel mean flow, and experimental determination of the effect of vibration amplitude and Mach number upon panel radiation efficiency.

77-875

Acoustoelasticity

E.H. Dowell

Princeton Univ., NJ, In: NASA, Langley Res. Ctr. Advan. in Eng. Sci., Vol 3, 1976, pp 1057-1070 (see N77-10305 01-31)
N77-10326

Key Words: Vibrating structures, Sound generation, Enclosures, Mathematical models

Internal sound fields are considered. Specifically, the interaction between the (acoustic) sound pressure field and the (elastic) flexible wall of an enclosure is discussed. Such problems frequently arise when the vibrating walls of a transportation vehicle induce a significant internal sound field. Cabin noise in various flight vehicles and the internal sound field in an automobile are representative examples. A mathematical model, simplified solutions, and numerical results and comparisons with representative experimental data are briefly considered.

77-876

The Assessment of Noise from Industrial Plants by Direct Measurement and by Calculation

M. Grashof

Bayer AG Leverkusen, IN Anlagenplanung Erdöl-chemie, c/o Erdölchemie GmbH, 5000 Köln 71, Postfach 752002, West Germany, Appl. Acoust., 9 (3), pp 177-192 (July 1976) 9 figs, 6 refs

Key Words: Industrial facilities, Noise generation

The noise levels measured at a distance of 1000m from an industrial plant can vary within a range of 20 dB(A) due to the effects of weather and extraneous sources. This paper examines the use of various statistical parameters for assessing the noise due to the plant and describes a method of minimizing the effect of extraneous noise due to traffic. The choice of parameter influences the attenuation data to be used in calculating the noise from new plants. Some typical measurements from a large industrial plant are presented. The paper emphasizes the need to agree on a single parameter which can be used for calculations, for assessing measured noise levels, and for comparison with legal or noise criteria, and recommends the general adoption of L_{eq}.

77-877

The Prediction and Measurement of Sound Radiated by Structures

R.H. Lyon and J.D. Brito

Massachusetts Inst. of Tech., Cambridge, MA., In: NASA, Langley Res. Ctr., Advan. in Eng. Sci., Vol. 3, 1976, pp 1031-1042 (see N77-10305 01-31)

N77-10324

Key Words: Sound waves, Machinery, Noise source identification, Noise reduction

Theories regarding the radiation of sound are reviewed and the implementation in strategies for explaining or measuring the sound produced by practical structures are discussed. Particular attention is given to those aspects that relate to the determination of the relative amounts of sound generated by various parts of a machine or structure, which can be very useful information for noise reduction efforts.

77-878

Noise Propagation in Urban and Industrial Areas

H.G. Davies

New Brunswick Univ., Fredericton, In: NASA, Langley Res. Ctr. Advan. in Eng. Sci., Vol. 3, 1976, pp 997-1008 (see N77-10305 01-31)

N77-10321

Key Words: Noise propagation, Motor vehicle noise

Noise propagation in streets and the discrepancies between theoretical analyses and field measurements are discussed. A cell-model is used to estimate the general background level of noise due to vehicular sources distributed over the urban area.

77-879

Traffic Noise Prediction with Particular Reference to Free-Flow Road Traffic

D. Brown

Dept. of Transport Tech., Loughborough Univ. of Tech., England, Rept. No. TT-7507, 41 pp (Aug 1975)

Sponsored by the Sci. Res. Council

N76-33724

Key Words: Noise prediction, Traffic noise, Mathematical models

Predictive equations for road traffic noise intensity and duration were derived from purely analytical considerations and are further developed by the inclusion of a noise peak level statistical model derived from published data. The resulting equations are expressed in terms of commonly used traffic parameters, and tested against measured data for a wide range of conditions.

77-880

A Model for the Prediction of Noise Levels Arising from Typical Airport Operations

F.K.I. Hirji and D.M. Waters

Dept. of Transport Tech., Loughborough Univ. of Tech., England, Rept. No. TT-7511, 130 pp (July 1975)

N76-33725

Key Words: Noise prediction, Aircraft noise, Airports, Mathematical models

A noise exposure prediction method requiring a minimum of information on aircraft movements was developed for airport noise. A standard case was defined which relates to a fully developed regional airport, the noise contours of which are presented. In order to make comparisons and combinations with other possible noise sources, the noise exposure was calculated in mean energy level units as well as using the conventional noise and number index. The new U.S. unified unit, the day/night mean energy level was also computed in an approximate manner.

RANDOM

(Also see No. 932)

77-881

Mean-Square Response of a Nonlinear System to Nonstationary Random Excitation

H. Kanematsu and W.A. Nash
Univ. of Massachusetts, Dept. of Civil Engrg., Amherst, MA 01002, Rept. No. AFOSR 72-2340, 58 pp (Aug 1976)

Key Words: Random vibrations, Nonlinear systems

The transient mean-square response of a nonlinear single degree of freedom mechanical system to nonstationary random excitation characterized by the product of an envelope function and a stationary Gaussian random process is determined by the equivalent linearization technique. A unit step envelope function is considered in conjunction with both correlated and white noise with zero mean. It has been shown that for white noise modulated by a unit step function, the transient mean-square response never exceeds the stationary response.

SEISMIC

(Also see Nos. 962, 982, 983, 1032)

77-882

Least Weight Structures for Threshold Frequencies in a Seismic Environment

R.D. McConnell
Structural Div., Veterans Administration, Washington D.C. 20420, Intl. J. Computers and Structures, 7 (1), pp 157-160 (Feb 1977) 1 fig, 28 refs

Key Words: Seismic design, Optimization, Substructure technique

Conventional component seismic design procedures fall into two categories: a table of component forces, or sophisticated computer analyses using detailed models which undergo dynamic time-history force simulations. Aerospace engineers have designed many component structures to a minimal-fundamental, or "threshold" frequency specification. Although it does not appear as generally applicable for ground-supported "base structures", the design of substructures for earthquake induced forces can similarly adopt such specifications.

SHOCK

(Also see Nos. 979, 996, 1012, 1016, 1017, 1018, 1019, 1020, 1021, 1022, 1023, 1024, 1027)

77-883

Symposium on Explosives and Pyrotechnics

Proc. of the 9th, Franklin Inst. of Tech., Philadelphia, PA, 32 papers, 576 pp (Sept 15-16, 1976)

Key Words: Pyrotechnic shock environment, Explosives, Proceedings

The papers, presented at four sessions, were concerned with basic studies, measuring techniques, testing and performance studies, new developments in the field, and safety techniques.

77-884

A Study of Convergence and Stability of Finite Element Approximation of Shock and Acceleration Waves in Nonlinear Materials

J.T. Oden, L.C. Wellford, Jr., and C.T. Reddy
Dept. of Aerospace Engrg. and Engrg. Mechanics, Texas Univ. at Austin, Austin, TX, Rept. No. ARO-11860.9-M, 305 pp (Aug 1976)
AD-A031 238/9GA

Key Words: Shock wave propagation, Finite element techniques

This report summarizes the recent work on the development of discontinuous finite element methods for the analysis of shock waves in nonlinear elastic materials. A class of one-dimensional finite elements is introduced in which the local interpolation functions consist of the usual piecewise linear functions and some additional functions which have discontinuities. In this way it is possible to model the local displacement field in terms of the values of the displacement at each node and two additional terms in which the shock strength and the location of the shock within an element are used as parameters. The corresponding variational formulation contains the required jump conditions. For a specific class of material a priori error estimates are derived and the scheme is implemented and applied to a number of representative examples.

PHENOMENOLOGY

COMPOSITE

(See No. 913)

DAMPING

(Also see Nos. 968, 1000)

77-885

Numerical Determination of the Transmissibility Characteristics of a Squeeze Film Damped Forced Vibration System

M.A. Sutton and P.K. Davis

In: NASA, Langley Res. Ctr., Advan. in Eng. Sci., Vol. 1, 1976, pp 327-338 (see N77-10230 01-31) N77-10262

Key Words: Squeeze film damping, Buildings, Seismic excitation, Mathematical models

Numerical solutions of the governing equations of motion of a liquid squeeze film damped forced vibration system were carried out to examine the feasibility of using a liquid squeeze film to cushion and protect large structures, such as buildings, located in areas of high seismic activity. The mathematical model used was that for a single degree of freedom squeeze film damped spring mass system. The input disturbance was simulated by curve fitting actual seismic data with an eleventh order Lagrangian polynomial technique. Only the normal component of the seismic input was considered. The nonlinear, nonhomogeneous governing differential equation of motion was solved numerically to determine the transmissibility over a wide range of physical parameters using a fourth-order Runge-Kutta technique. It is determined that a liquid squeeze film used as a damping agent in a spring-mass system can significantly reduce the response amplitude for a seismic input disturbance.

77-886

Design of Nonlinear Squeeze-Film Dampers for Aircraft Engines

E.J. Gunter, L.E. Barrett,and P.E. Allaire

Dept. of Mech. Engrg., Univ. of Virginia, Charlottesville, VA., J. Lubric. Tech., Trans. ASME, 99 (1), pp 57-64 (Jan 1977) 12 figs, 10 refs

Sponsored by NASA

Key Words: Squeeze film damping, Squeeze film bearings, Aircraft engines

This paper examines the effect of squeeze-film damper bearings on the steady state and transient unbalance response of aircraft engine rotors. The nonlinear effects of the damper are examined, and the variance of the motion due to unbalance, static pressurization, retainer springs, and rotor preload is shown.

77-887

Transmissibility Study of a Flexibly Mounted Rolling Element Bearing in a Porous Bearing Squeeze-Film Damper

C. Cusano and P.E. Funk

Dept. of Mech. and Industrial Engrg., Univ. of Illinois at Urbana-Champaign, Urbana, IL, J. Lubric. Tech., Trans. ASME, 99 (1), pp 50-56 (Jan 1977) 10 figs, 11 refs

Key Words: Squeeze film damping, Squeeze film bearings, Aircraft engines

The purpose of the investigation is to study the transmissibility characteristics of a centrally preloaded porous bearing squeeze-film damper supporting a rolling element bearing. The data presented are for squeeze-film porous bearings having a wall thickness-to-length ratio of 0.1 and three degrees of permeability, including the case of zero permeability which corresponds to a solid bearing.

77-888

A Decade of Damping

E.E. Ungar

Bolt Beranek and Newman Inc., Cambridge, MA

S/V, Sound Vib., 11 (1), p 11 (Jan 1977)

Key Words: Damping, Reviews

The terminology and technology of damping, resulting from recent published works, are reviewed.

FATIGUE

77-889

Accelerated Vibration Fatigue Life Testing of Leads and Soldered Joints

H.S. Blanks

School of Electrical Engrg., Univ. of New South Wales, Sydney, Australia, Microelectronics and Reliability, 15, pp 213-219 (1976) 4 figs, 7 refs

Key Words: Fatigue tests

The prediction of vibration fatigue life of leads and solder joints at low excitation from that obtained during high-level testing is possible but the extrapolation exponent depends not only on the S-N curve of the failed material, but also on the type of excitation and on whether failure is due to component or to support resonance. Varied materials and excitation types were studied.

77-890

Digital Simulation of Environmental Processes with Respect to Fatigue

M. Bily and J. Bukoveczky
Inst. of Machine Mechanics of the Slovak Academy of Sciences, Bratislava-Patronka, Czechoslovakia, J. Sound Vib., 49 (4), pp 551-568 (1976) 2 figs, 11 refs

Key Words: Fatigue life, Digital simulation

This paper presents various methods of on-line digital simulation of real environmental processes, representing inputs of fatigue machines. According to the method of analysis adopted, this simulation is based either on sinusoidal amplitudes in the form of blocks or individual cycles, or on random processes. The random process simulation reproduces the probability density of amplitudes, the power spectral density, or both of them, depending on the fatigue damage hypothesis and philosophy of testing accepted.

FLUID

(Also see Nos. 930, 940, 941, 942, 963)

77-891

Aeroelastic Response of Square and H-Sections in Turbulent Flows

J. Tai, C.T. Crowe, and J.A. Roberson
D.W. Thompson Consultants, Ltd., Vancouver, B.C. Canada, ASME Paper No. 76-WA/FE-19

Key Words: Fluid-induced excitation, H-beams

The aeroelastic response of the square and H-section in a turbulent cross-flow was investigated experimentally. At low turbulence-intensity levels (1 percent) the square-section response appears to be a vortex-induced oscillation superimposed on a galloping oscillation, while at high turbulence intensities (10 percent) the vortex-induced oscillation is not apparent. When the flange of the H-section is normal to the flow, the response is the classic vortex-induced oscillation and is affected little by turbulence intensity. When the flange is parallel to the flow, the response can be characterized as vortex-induced at low turbulence intensities and galloping at high turbulence intensities, while at intermediate intensities (5 percent), the response appears to be a composite effect of both mechanisms. The inability to explain the observed trends demonstrates the need for more data on the aeroelastic behavior of these important structural shapes.

SOIL

(See No. 966)

EXPERIMENTATION

DIAGNOSTICS

77-892

Experiences with a Minicomputer-Based Machinery Monitoring System

C. Bultzo
Exxon Co., Baytown, TX, ASME Paper No. 76-Pet-11

Key Words: Diagnostic techniques, Machinery, Computer-aided techniques

A real-time monitoring system that utilizes existing data-acquisitioning hardware, a mini-computer, a new signal-conditioning hardware, and software is described. Included are comments on the objectives of the system, what is monitored, monetary justification, and "growing pains."

77-893

Vibration Energy: A Quick Approach to Rotor Dynamic Optimization

H.R. Simmons
Southwest Research Institute, San Antonio, TX, ASME Paper No. 76-Pet-60

Key Words: Rotors, Vibration response, Optimization, Machine diagnostics

A method is described for identifying the most significant rotor parts contributing to a vibration problem and for calculating critical speed changes due to structural modification of those parts. Thus, this technqiue is highly suitable for troubleshooting and correcting field vibration problems.

77-894

A Dedicated Minicomputer Based Monitoring System for Turbomachinery Protection and Analysis

R.G. Harker

Bently Nevada Corp., Minden, NV, ASME Paper No. 76-Pet-13

Key Words: Diagnostic techniques, Turbomachinery, Computer-aided techniques

This paper describes the application of a minicomputer for machinery monitoring, diagnostics, analysis, and evaluation of performance. The functions of various computer subsystems are shown in relation to the entire system.

77-895

Rotor Dynamics Analysis Instrument Introduced

Diesel and Gas Turbine Progress, 18 (1), pp 66-67 (Jan 1977)

Key Words: Diagnostic instrumentation, Balancing machines, Rotating structures, Turbomachinery, Pumps, Turbines, Fans, Compressors

This article describes Bently Nevada Corporations's latest rotor dynamics data collection system. Known as the DVF 2, this computer-compatible instrument incorporates all of the capabilities and features of Bently Nevada's original digital vector filter with added functions, flexibility, applications, and performance improvements. The DVF 2 is an advanced design data collection tool for diagnosing and balancing virtually all rotating equipment, and typcially: steam turbines, gas turbines, electric motors, centrifugal compressors, reciprocating compressors, screw compressors, generators, hydroelectric turbines, pumps, gears, fans, and centrifuges.

77-896

Vibration Condition Monitoring Techniques for Rotating Machinery

B. Dawson

Div. of Engrg., The Polytechnic of Central London, London W1M 8JS, England, Shock Vib. Dig., 8 (12), pp 3-8 (Dec 1976) 41 refs

Key Words: Diagnostic techniques, Rotating structures, Bearings, Gears, Reviews

This paper reviews vibration monitoring techniques currently available for rotating machinery. Applications of these techniques to bearings and gears are described.

77-897

Gear Case Distortion Causes Failures

F.L. VanLaningham

Union Carbide Corp., South Charleston, WV, Hydrocarbon Processing, 56 (1), pp 91-96 (Jan 1977) 7 figs

Key Words: Gear boxes, Diagnostic techniques

Some diagnostic techniques which apply to vibration problems caused by gear case distortion are described.

77-898

Qualify Vessel Integrity with Acoustic Emission Analysis

D.L. Parry

Exxon Nuclear Co., Inc., Richland, WA, Hydrocarbon Processing, 55 (12), pp 132-134 (Dec 1976) 2 figs

Key Words: Diagnostic techniques, Acoustic techniques, Nondestructive testing, Containers

A nondestructive testing tool to qualify structural integrity of pressure containment systems in nuclear, petrochemical, and pipe line industries which uses the emission of minute pulses of elastic energy in materials under stress is described. The pulse, when properly detected and analyzed, provide information on the discontinuities of the structure under stress.

77-899

Development of a Diagnostic System for Off-Highway Vehicle Power Trains

J. L. Frarey and R.F. Burchill

Shaker Research Corp., Ballston Lake, NY, 59 pp (Oct 1976) (see also AD-777 811)
AD-A031 627/3GA

Key Words: Diagnostic instrumentation, Off-highway vehicles, Tractors, Power transmission systems

This report deals with an investigation on the commonality of high frequency vibration data from a D8 and a smaller crawler tractor the D7. Based on the data from the D8, a spectral region between 20 KHz and 50 KHz was selected. An engineering model of a diagnostic system was built and tested that will operate on both the Caterpillar D7 and D8 crawler tractors.

77-900

Progress and Payout of a Machinery Surveillance and Diagnostic Program

R.J. Hudachek and V.R. Dodd

Standard Oil Co., Pascagoula, MS, ASME Paper No. 76-Pet-69

Key Words: Diagnostic instrumentation

This paper describes and economically justifies new procedures to obtain maximum utilization of advanced vibration monitoring and analysis equipment as an alternate to high maintenance costs and reduced plant production.

EXPERIMENT DESIGN

(See No. 919)

FACILITIES

(Also see No. 862)

77-901

A New Wave-Wind Channel for Fluid Dynamics Research at the Naval Research Laboratory

O.M. Griffin and J.W. Wright

Naval Research Lab., Washington D.C., Rept. No. NRL-MR-3352, 22 pp (Aug 1976)

AD-A031 318/9GA

Key Words: Test facilities, Wind-induced excitation, Water waves

A new multipurpose wave-wind channel facility has been constructed for use in the support of on-going fluid dynamics research programs at the Naval Research Laboratory. This facility has been designed to accommodate experiments ranging from the study of wind and wave effects on ocean waves. The channel has been equipped initially with a mechanically-driven regular wave generator and a wind tunnel for the generation of wind waves. The laboratory facility, the wave channel and wave generators, and the research programs for which the laboratory has been constructed are described in this report.

77-902

Development of an Acoustic Facility for Determination of Sound Power Emitted by Appliances

R.P. Harmon

Underwriters Laboratories, Inc., Northbrook, IL, Noise Control Engr., 7 (2), pp 110-114 (Sept/Oct 1976) 7 figs, 3 refs

Key Words: Test facilities, Household appliances, Noise measurement

The subject of this paper is a prefabricated facility developed for determination of sound power emitted by appliances. It consists of a reverberant room, microphone traverse, and real time analysis system interfaced with a programmable calculator.

INSTRUMENTATION

(Also see Nos. 899, 900)

77-903

Sound and Vibration Measuring Instrumentation

G.W. Kamperman

S/V, Sound Vib., 11 (1), pp 8-9 (Jan 1977)

Key Words: Measuring instruments, Sound measurement, Vibration measurement, Reviews

Some of the highlights on sound and vibration instrumentation that have taken place during the past decade are described. This information is extrapolated to predict advances during the next decade.

77-904

Shock and Vibration Instrumentation

R. Plunkett

Dept. of Aeronautics and Engineering Mechanics, Univ. of Minnesota, Minneapolis, MN 55455, Shock Vib. Dig., 8 (12), pp 21-26 (Dec 1976) 87 refs

Key Words: Shock measurement, Vibration measurement, Measuring instruments, Reviews

This article briefly describes recent developments in instrumentation used to measure shock and vibration.

77-905

Instrumentation for Measuring Aircraft Noise and Sonic Boom

A.J. Zuckerwar

NASA, Langley Res. Ctr., Langley Station, VA., U.S. Patent-3 964 319, 8 pp (June 1976)

Sponsored by NASA

Key Words: Measuring instruments, Aircraft noise, Sonic boom

Improved instrumentation suitable for measuring aircraft noise and sonic booms is described. An electric current proportional to the sound pressure level at a condenser microphone is produced and transmitted over a cable and amplified by a zero drive amplifier.

77-906

New Optical Method to Determine Vibration-Induced Strains with Variable Sensitivity After Recording

Y.Y. Hung, J.D. Hovanesian and A.J. Durelli
School of Engrg., Oakland Univ., Rochester, MI.,
Rept. No. 41, 26 pp (Nov 1976)
AD-A032 087/9GA

Key Words: Optical measuring instruments, Vibration effects

A steady-state vibrating object is illuminated with coherent light and its image is slightly misfocused in the film plane of a camera. The resulting processed film is called a 'Time-integrated specklegram.' When the specklegram is Fourier filtered, it exhibits fringes depicting derivatives of the vibrational amplitude. The direction of the spatial derivative, as well as the fringe sensitivity may be easily and continuously varied during Fourier filtering process.

77-907

Apparatus and Method for the Remote Detection of Vibrations of Diffuse Surfaces

V.J. Rosati
Dept. of the Army, Washington, D.C.,
U.S. PATENT-3 952 583, 4 pp (Apr 27, 1976)

Key Words: Vibration detectors

Apparatus and method for the remote detection of vibrations of diffuse surfaces is described. A beam of coherent light incident on a vibrating diffuse surface is scattered. A photodetector sees the scattering as a time-varying intensity distribution related to the vibration frequency of the surface. The detected time-varying intensity distribution is converted into time-varying electrical currents which can be rendered audible by using a loudspeaker or visual by using an oscilloscope. The invention provides a novel method and apparatus for the covert surveillance of distant conversations, movements of people, movements of vehicles, and the like.

TECHNIQUES

(Also see Nos. 906, 907)

77-908

On the Use of Acoustical Holography to Locate Sound Sources on Complex Structures

W.F. King and E.E. Watson
Inst. fuer Turbulenzforschung, Deutsche Forschungs- und Versuchsanstalt fuer Luft- und Raumfahrt, Berlin, West Germany,
Rept. No. DLR-FB-76-12, 35 pp (Apr 12, 1976)
Sponsored by the Naval Sea System Command
N77-10870

Key Words: Acoustic holography, Noise source identification, Fans, Ventilation

An experimental technique is presented for the identification of far field sound sources on complex vibrating structures. Results are given for a preliminary investigation designed to determine the feasibility of applying a modified form of acoustic holography to the location of sound sources on a three-bladed ventilator fan. Optical and computer sound source reconstructions are compared and analyzed.

77-909

Time-Averaged Shadow-Moire Method for Studying Vibrations

Y. Y. Hung, C. Y. Liang, J. D. Hovanesian and A.J. Durelli
School of Engrg., Oakland Univ., Rochester, MI,
Rept. No. 39, 22 pp (Nov 1976)
AD-A032 081/2GA

Key Words: Photographic techniques, Vibration effects, Moire effects, Large amplitudes

A time-averaged shadow-moire method is presented which permits the determination of the amplitude distribution of the deflection of a plate in steady state vibration. No stroboscope is required and the recording is done statically.

COMPONENTS

ABSORBERS

(See Nos. 932, 933, 934)

BEAMS, STRINGS, RODS

(Also see Nos. 859, 864, 866, 891, 968, 1007)

77-910

Prediction of the Angular Vibration of Aircraft Structures

J. Lee and P.W. Whaley

AF Flight Dynamics Lab., Wright-Patterson Air Force Base, OH 45433, J. Sound Vib., 49 (4), pp 541-549 (1976) 2 figs, 11 refs

Key Words: Aircraft, Beams, Rotatory inertia effects, Bernoulli-Euler method

With the Bernoulli-Euler beam used as a theoretical model, qualitative relationships have been developed to predict the angular vibration from a prescribed linear vibration response. Based on laboratory and flight test data, the overall accuracy of predictions is within ±20% of the angular vibration measurement.

77-911

On the Dynamic Response of a Prestressed Beam

A.D. Kerr

Dept. of Civil Engrg., Princeton Univ., Princeton, NJ 08540, J. Sound Vib., 49 (4), pp 569-573 (1976) 4 figs, 5 refs

Key Words: Beams, Prestressed structures, Natural frequencies

The dynamic bending response of a prestressed beam, in which the prestressing rod is continuously supported, is studied analytically and experimentally.

77-912

Performance of Viscoelastically Damped Multilayer Structures Subjected to Shock Excitation

A.D. Kapur and B.C. Nakra

Indian Inst. of Tech., New Delhi, India, 38 pp (Mar 1976) (Backup document for AIAA Synoptic scheduled for publication in AIAA J. in Jan 1977)
N76-32580

Key Words: Beams, Laminates, Damped structures, Viscoelastic properties

The performance of 5-layer, 3-layer, and 2-layer damped beams subjected to shock excitation was studied. The analysis was based on the effects of rotary and longitudinal inertias of the beam besides transverse inertia, and employs a 4-element viscoelastic model to describe the properties of the viscoelastic material in shear or direct strains.

77-913

An Analytical Method for Evaluating Impact Damage Energy of Laminated Composites

C.T. Sun

School of Aero. and Astro., Purdue Univ., Lafayette, IN, Rept. No. AFML-TR-76-87, 54 pp (June 1975)
AD-A030 781/9GA

Key Words: Beams, Laminates, Impact shock, Finite element technique

A higher order beam finite element is developed for dynamic response of beams subjected to impact of elastic spheres. Hertzian law is used to evaluate the contact force.

77-914

Effect of Longitudinal or Inplane Deformation and Inertia on the Large Amplitude Flexural Vibrations of Slender Beams and Thin Plates

I.S. Raju, G.V. Rao, and K.K. Raju

Vikram Sarabhai Space Centre, Trivandrum-695022, India, J. Sound Vib., 49 (3), pp 415-422 (1976) 11 refs

Key Words: Beams, Plates, Circular plates, Rectangular plates, Flexural vibration

Large amplitude flexural vibrations of slender beams, and thin circular and rectangular plates have been studied when a compatible longitudinal or inplane mode is coupled with the fundamental flexural mode.

77-915

Higher Modal Dynamic Plastic Behavior of Beams Loaded Impulsively

N. Jones and C.A.P. Guedes Soares
Dept. of Ocean Engrg., Massachusetts Inst. of Tech., Cambridge, MA, Rept. No. 76-5, 56 pp (Oct 1976)
AD-A031 807/1GA

Key Words: Beams, Modal analysis, Plastic properties

The higher modal dynamic plastic response of fully clamped beams has been examined using various rigid perfectly plastic theoretical procedures and a numerical elastic-plastic computer code.

77-916

A Study of the Higher Modal Dynamic Plastic Response of Beams

N. Jones and T. Wierzbicki
Dept. of Ocean Engrg., Massachusetts Inst. of Tech., Cambridge, MA 02139, Intl. J. Mech. Sci., 18, pp 533-542 (Nov/Dec 1976) 8 refs

Key Words: Beams, Modal analysis

A theoretical and experimental investigation into the higher modal dynamic plastic response of fully clamped beams is reported herein.

77-917

Sound Radiation from Randomly Vibrating Beams of Finite Circular Cross Section

M.W. Sutterlin and A.D. Pierce
Bolt Beranek and Newman Inc., Cambridge, MA In: NASA, Langley Res. Ctr. Advances in Engrg. Sci., Vol. 3, 1976, pp 1071-1081 (See N77-10305 01-31)
N77-10327

Key Words: Beams, Vibrating structures, Sound generation, Textile looms

The radiation of sound from vibrating cylindrical beams is analyzed based on the frequency of the beam vibrations and the physical characteristics of the beam and its surroundings. A statistical analysis of random beam vibrations allows this result to be independent of the boundary conditions at the ends of the beam. The acoustic power radiated by the beam can be determined from a knowledge of the frequency band vibration amplitudes. A practical example of the usefulness of this technique is provided by the application of the theoretical calculations to the prediction of the octave band acoustic power output of the picking sticks of an automatic textile loom. Calculations are made of the expected octave band sound pressure levels based on measured acceleration data. These theoretical levels are subsequently compared with actual sound pressure level measurements of loom noise.

77-918

On Wave Solutions in Overhead Wire Dynamics

E.N. Fox
Engrg. Dept., Univ. of Cambridge, Cambridge, CB2 1PZ England, Intl. J. Mech. Sci., 18, pp 417-429 (July/Aug 1976) 1 fig, 6 refs

Key Words: Catenaries, Transmission lines, Moving loads

This paper is concerned with the dynamics of the 'simple catenary' type of overhead contact system for current collection on electrified railway lines. A wave solution is derived for the dynamic response of the overhead system to a time-dependent travelling concentrated load applied direct to the wire. A wave solution is derived for a special case of loading through a rigid mass travelling in point contact with the wire.

77-919

An Experimental Study of Flow-Induced Motions of Flexible Cables and Cylinders Aligned with the Flow Direction

R.J. Hansen and C.C. Ni
Naval Research Lab., Washington, D.C., ASME Paper No. 76-WA/FE-15

Key Words: Cylinders, Cables, Fluid-induced excitation, Photographic techniques

Photographic studies of the flow-induced motions of flexible cables and cylinders aligned with the mean flow direction are reported. The experiments are conducted in a 7½-in.-dia test section which has a fully developed annular flow at the upstream end of the flexible member. The experimental results establish the dependence of the flow-induced motions on the flow velocity, tension in the cylinder, and roughness of the cylinder surfaces as well as on the configuration at the downstream end.

BEARINGS

(Also see Nos. 886, 887, 896)

77-920

The Dynamic Behaviour of Passively Compensated, Hydrostatic Journal Bearings with Various Numbers of Recesses

P.B. Davies

Dept. of Aero. and Mech. Engrg., Univ. of Salford, England, J. Mech. Engr. Sci., 18 (6), pp 292-300 (1976) 4 figs, 34 refs

Key Words: Journal bearings

A previously established small-perturbation analysis is developed to express the unsteady-state continuity-of-flow equation for an isolated recess in a passively compensated, multirecess, hydrostatic journal bearing in terms of generalized co-ordinates. The concise form of this equation enables motion of the shaft about the concentric position to be described by equations which are derived in closed form for bearings with orifice, capillary or constant flow compensation and any number or recesses. The results of the analysis, which is exact within the stated assumptions, are compared with those of other workers and the steady-state solution of the equations of motion is shown to give an expression for static stiffness which is useful for design purposes. Numerical values of the dynamic constant for bearings with between 3 and 20 recesses are given graphically.

77-921

The Fluid Pivot Journal Bearing

D.V. Nelson and L.W. Hollingsworth

General Electric Co., Sunnyvale, CA, J. Lubric. Tech., Trans. ASME, 99 (1), pp 122-128 (Jan 1977) 11 figs, 4 refs

Key Words: Journal bearings, Dynamic response

A promising new type of tilting pad journal bearing -- the Fluid Pivot journal bearing -- is described. The long development history of this bearing is summarized, showing how the design evolved through research and testing. Static and dynamic performance features of the bearing are presented, with sample test results compared with predictions based on an advanced computer analysis. Comparisons of the Fluid Pivot journal bearing with conventional mechanically pivoted journal bearings are given.

77-922

Dynamic Stiffness and Damping of Externally Pressurized Gas Lubricated Journal Bearings

D.P. Fleming, W.J. Thayer, and R.E. Cunningham

Lewis Research Ctr., Cleveland, OH, J. Lubric. Tech., Trans. ASME, 99 (1), pp 101-105 (Jan 1977) 6 figs, 8 refs

Key Words: Journal bearings, Gas bearings, Damping coefficients, Stiffness coefficients

A rigid vertical shaft was operated with known amounts of unbalance at speeds up to 30,000 rpm and gas supply pressure ratios to 4.8. From measured amplitude and phase angle data, dynamic stiffness and damping coefficients of the bearings were determined. The measured stiffness was proportional to the supply pressure, while damping was little affected by supply pressure. Damping dropped rapidly as the fractional frequency whirl threshold was approached.

77-923

A Theoretical Analysis of a Compliant Shell Air Bearing

K.P. Oh and S.M. Rohde

Engrg. Mech. Dept., Research Laboratories, General Motors Corp., Warren, MI, J. Lubric. Tech., Trans. ASME, 99 (1), pp 75-81 (Jan 1977) 5 figs, 13 refs

Key Words: Journal bearings, Gas bearings, Shells, Finite element technique

A mathematical analysis of a class of compliant journal bearings operating with a compressible lubricant is presented. Finite element methods are used to obtain approximate solutions, and results for the rigid case are compared to those obtained by finite difference methods. Results for the case in which the compliant bearing component is a shell show that the steady-state performance can be radically different from that predicted by the rigid bearing assumption. These differences may explain the antiwhirl characteristics exhibited by this type of bearing.

77-924

Research Note: Film Rupture in a Dynamically-Loaded Non-Conformal Contact

D. Dowson, E.H. Smith, and C.M. Taylor

Inst. of Tribology, The Univ. of Leeds, England, J. Mech. Engr. Sci., 18 (6), pp 303-305 (1976) 3 figs, 11 refs

Key Words: Bearings, Rotor-bearing systems, Lubrication

This note is concerned with the rupture of hydrodynamically lubricated films in infinitely-wide non-conformal contacts. An attempt to correlate previous experimental work with new analytical results using separation cavitation boundary conditions is reported.

77-925

A Study of the Lubrication and Dynamics of Geometrically Perfect, Lightly Loaded, Cylindrical Roller Bearings with Particular Reference to Shaft Loci

P.H. Markho and D. Dowson

Dept. of Mech., Marine and Production Engrg., Liverpool Polytechnic, England, J. Mech. Engr. Sci., 18 (6), pp 263-270 (Dec 1976) 10 figs, 12 refs

Key Words: Rotor-bearing systems, Lubrication

The principal objective of this paper is to ascertain and quantify the cyclic movement of the centre of a shaft in a geometrically perfect, lubricated, cylindrical roller bearing exhibiting initial clearance and subjected to a light and steady load. The movement may be important in relation to the accuracy of location of shafts in roller bearings. The study also covers more conventional features of bearing performance such as lubricant film thickness and co-efficient of friction, and shows the effect of squeeze to be negligible under steady bearing loads.

77-926

Static and Dynamic Performance of an Infinite Stiffness Hydrostatic Thrust Bearing

N. Tully

Dept. of Mech. Engrg., Univ. of Natal, South Africa, J. Lubric. Tech., Trans. ASME, 99 (1), pp 106-112 (Jan 1977) 7 figs, 5 refs

Key Words: Thrust bearings, Dynamic response

A novel form of variable hydrostatic restriction is proposed which will automatically achieve a high, infinite or negative static stiffness over a substantial load range. The restrictor is formed between the bearing body and a spring mounted conical plug. The steady state performance is analyzed and design curves presented which are valid for any cone angle from zero, i.e., fixed clearance, to 90 deg which is the normal diaphragm restrictor. The dynamic response to forced sinusoidal vibrations is examined in conventional vibration analysis form and it is found that the restrictor system may be designed to act as a vibration absorber.

BLADES

(Also see Nos. 869, 992, 1007)

77-927

Control of Rotary Lawn Mower Noise

D.A. Guenther, M.J. Moran, and L.L. Faulkner

Dept. of Mech. Engrg., The Ohio State Univ., Columbus, OH, Appl. Acoust., 10 (1), pp 9-18 (Jan 1977) 3 figs, 30 refs

Key Words: Lawn mowers, Rotor blades, Noise reduction

The current state of knowledge regarding the noise generated by rotary lawn mowers is reviewed, with emphasis on the role of the rotating blade. Noise control methods are discussed and evaluated.

77-928

Research Note: Effect of Mistuning on Forced Vibration of Blades with Mechanical Coupling

D.S. Whitehead

Engrg. Dept., Univ. of Cambridge, England, J. Mech. Engr. Sci., 18 (6), pp 306-307 (1976) 3 refs

Key Words: Coupled systems, Blades, Forced vibration

The maximum factor by which the vibration amplitude can increase due to mistuning when there is mechanical coupling between blades through their root fixings is given.

77-929

A Finite Element Analysis for the Vibration Modes of a Bladed Disc

J. Kirkhope and G.J. Wilson

Dept. of Mech. and Aero. Engrg., Carleton Univ., Ottawa, Canada K1S 5B6, J. Sound Vib., 49 (4), pp 469-482 (Dec 22, 1976) 6 figs, 27 refs

Key Words: Blades, Discs, Coupled response, Finite element technique

The coupled vibration modes of a rotating blade-disc system are calculated by a finite element method. It is assumed that a large number of identical blades are present, such that the resulting blade loadings on the disc can be considered continuously distributed around the rim of the disc. The disc may have arbitrary profile, and the blades may be tapered and twisted, thus closely representing practical axial flow turbomachine configurations. The effects of rotation, thermal stress, and transverse shear and rotatory inertia in discs of moderately thick profile are readily incorporated into the finite element model. Calculated values of frequencies are compared with experimental data obtained on non-rotating models, and the convergence of the solution is examined by comparison with exact solutions, which can be obtained for configurations of simple geometry.

CYLINDERS

(Also see No. 919)

77-930

Added Mass and Damping Forces on Circular Cylinders

R.A. Skop, S.E. Ramberg, and K.M. Ferer

Naval Research Lab., Washington, D.C., ASME Paper No. 76-Pet-3

Key Words: Circular cylinders, Floating structures, Viscous damping

A series of experiments has been performed to determine the fluid added mass and damping forces on harmonically oscillating cylinders in still water. The forces obtained in this case are simply related to the forces which would occur if the cylinder was stationary and the fluid oscillated with the same amplitude and frequency. Hence, the results have direct bearing on the calculation of wave forces on cylindrical structures. Primary attention is given to low amplitude motions where it is found that the fluid damping force is purely viscous (proportional to the velocity).

77-931

Vibrations and Stresses in Layered Anisotropic Cylinders

G.P. Mulholland and B.P. Gupta

New Mexico State Univ., University Park, NM, In: NASA Langley Res. Center Advances in Engrg. Sci., Vol. 2, 1976, pp 459-472 (See N77-10265) N77-10273

Key Words: Cylindrical bodies, Anisotropy, Laminates, Vibration response

An equation describing the radial displacement in a k layered anisotropic cylinder was obtained. The cylinders are initially unstressed but are subjected to either a time dependent normal stress or a displacement at the external boundaries of the laminate. Numerical examples are given to illustrate the procedure.

DUCTS

(Also see Nos. 932, 969, 975)

77-932

Experimental Verification of a Finite Length Tuning Concept for Acoustic Lining Design

J.F. Unruh and I.R. Price

Boeing Commercial Airplane Co., Seattle, WA 98124, J. Sound Vib., 49 (3), pp 393-402 (1976), 15 figs, 3 refs

Key Words: Ducts, Acoustic linings, Noise reduction, Hole-containing media

A program was carried out to experimentally verify an acoustic finite length tuning concept for low frequency lining designs. By properly accounting for the reflective properties of discontinuous admittance via mode matching techniques, five panels were designed to demonstrate the length tuning concept. Three panels were designed for no flow and two panels were designed for a flow Mach number of 0-2. The panels were of conventional design with perforated face sheets and rectangular cavity resonators.

77-933

Noise Generated by Boundary-Layer Interaction with Perforated Acoustic Liners

A.B. Bauer and R.L. Chapkis

Douglas Aircraft Co., Long Beach, CA 90800, J. Aircraft, 14 (2), pp 157-160 (Feb 1977) 5 figs, 19 refs

Key Words: Ducts, Acoustic linings, Hole-containing media, Noise generation

A problem that occurs in the application of perforated plate acoustic duct liners is the noise generated by the turbulent boundary-layer flow over the holes in the liner surface. This flow not only generates noise but also thickens the boundary layer. To observe the noise generation, a series of tests have been run.

77-934

Sound Attenuation in Multiply Lined Rectangular Ducts Including the Effects of the Wall Impedance Discontinuities. Part 1: Liners in Series

W. Koch

Deutsche Forschungs- und Versuchsanstalt fuer Luft- und Raumfahrt, Goettingen, W. Germany, Rept. No. DLR-FB-76-07, 113 pp (Jan 15, 1976)

N77-10871

Key Words: Ducts, Acoustic linings, Noise reduction

An exact solution for the computation of sound attenuation in multiply lined ducts at zero convection velocity including the effects of the wall impedance discontinuities is given by means of an extended Wiener-Hopf method. This solution is evaluated numerically for a realistic impedance model. For the fundamental mode the influence of the different parameters is demonstrated in several diagrams for one- and two-layer, one- and two-part liners in series.

GEARS

(See No. 896)

LINKAGES

77-935

Resonance in Planar Linkage Mechanisms Mounted on Vibrating Foundations

B.S. Thompson and R.P. Ashworth

Dept. of Mech. Engrg., Univ. of Dundee, Dundee DD1 4HN, Scotland, J. Sound Vib., 49 (3), pp 403-414 (1976) 5 figs, 23 refs

Key Words: Linkages, Elastic foundations, Vibrating foundations

An investigation into the dynamic response of the elastic connecting rod of a slider crank mechanism whose foundation is subjected to a vibration perpendicular to the plane of the linkage is presented.

77-936

A New Torsional Elastic Seal for Oscillatory Motion

R.L. Orndorff, Jr. and J.H. Kramer

B. F. Goodrich Engineered Systems Co., Akron, OH, ASME Paper No. 76-Pet-64

Key Words: Rotary seals, Elastomers

A new seal is described, a number of applications are discussed, and the design procedure for a typical rubber element is given. Torsilastic seals are primarily intended for rotary and linear oscillating applications, but can be used for slow-speed continuous rotation applications where some or all of the load is carried by bearings and the rubber installation compressive force is low. The seal is a zero-leakage, fail-safe device that makes use of a Teflon-lined rubbing surface to limit the seal torque to a manageable value. Small angle oscillations are handled by torsional shear of the rubber, large angles by slip on the Teflon.

MEMBRANES, FILMS, AND WEBS

77-937

On the Explicit Finite Element Formulation of the Dynamic Contact Problem of Hyperelastic Membranes

J.O. Hallquist and W.W. Feng

Lawrence Livermore Lab., California Univ., Livermore, CA 94550, In: NASA, Langley Res. Center Advances in Engrg. Sci., Vol. 2 1976, pp 417-424 (See N77-10265 01-31)

N77-10269

Key Words: Membranes, Dynamic response, Finite element technique

Contact-impact problems involving finite deformation axisymmetric membranes are solved by the finite element method with explicit time integration. The formulation of the membrane element and the contact constraint conditions are discussed. The hyperelastic, compressible Blatz and Ko material is used to model the material properties of the membrane. Two example problems are presented.

77-938

Transmission of Free Waves Across a Rib on a Panel

G. Maidanik, A.J. Tucker, and W.H. Vogel

David W. Taylor Naval Ship Research and Development Center, Bethesda, MD 20084, J. Sound Vib., 49 (4), pp 445-452 (1976) 2 figs, 5 refs

Key Words: Membranes, Panels, Ribs (supports), Fluid-induced excitation

The transmission of free waves across a rib is investigated on a singly ribbed panel. The panel is in the form of a membrane that is made to possess dispersive properties similar to those of a thin plate. Of particular interest in the investigation is the influence that fluid loading and compliant coating may have on the transmission.

77-939

Free Vibration Analysis of a Composite Rectangular Membrane Consisting of Strips

K. Sato

Dept. of Mech. Engrg., Tohoku Univ., Sendai, Japan,
J. Sound Vib., 49 (4), pp 535-540 (Dec 22, 1976) 1 fig, 10 refs

Key Words: Rectangular membranes, Composite structures, Free vibration

The work is concerned with the free vibration problem of a composite rectangular membrane consisting of strips of different materials under certain initial conditions. Discussion of the special case of a homogeneous membrane is included.

PIPES AND TUBES

77-940

A Survey of Flow Induced Vibrations of Cylindrical Arrays in Cross-Flow

S.D. Savkar

Research and Development Ctr., General Electric Corp., Schenectady, NY, ASME Paper No. 76-WA/FE-21

Key Words: Tubes, Cylinders, Fluid-induced excitation

A phenomenological survey of flow induced vibrations of arrays of cylinders subjected to cross-flow is presented. The current state of empirical knowledge on added mass, vortex shedding, turbulent flow buffeting, and motion dependent instabilities is examined. The survey is directed to power plant components, such as steam generators, moisture-separator-reheaters, and other similar components.

77-941

Dynamics of Heat Exchanger Tube Banks

S.S. Chen

Argonne National Lab., Argonne, IL, ASME Paper No. 76-WA/FE-28

Key Words: Tubes, Heat exchangers, Fluid-induced excitation, Interaction: structure-fluid, Free vibrations, Forced vibrations

Flow-induced vibration in heat exchanger tube banks is of great concern, particularly in high performance heat exchangers used in nuclear reactor systems. In this paper, the dynamic characteristics of tube banks in stationary liquid are studied. A method of analysis is presented for free and forced vibrations of tube banks including tube/fluid interaction. Numerical results are given for tube banks subjected to various types of excitations.

77-942

Fluid Elastic Whirling of Tube Rows and Tube Arrays

R.D. Blevins

Southwestern Industries, Inc., Los Angeles, CA, ASME Paper No. 76-WA/FE-26

Key Words: Tubes, Fluid induced excitation, Whirling

Models are developed for the fluid force coefficients that determine the onset of whirling of tube rows and tube arrays. A control volume momentum analysis is employed. The results are in agreement with the available experimental data.

77-943

Dynamic Response Approximation for Noncircular Fluid Lines

M.E. Franke and E.F. Moore

Air Force Institute of Technology, Wright-Patterson Air Force Base, OH, J. Dyn. Syst. Meas. and Control, Trans. ASME, 98 (4), pp 421-424 (Dec 1976) 9 refs

Key Words: Pneumatic lines, Dynamic response, Fluid-induced excitation

Relatively simple methods are described for determining the approximate response of noncircular lines in terms of equivalent circular lines. This leads to the possible advantage that circular line equations and results can be used to predict the response of noncircular lines. Examples are given which compare previously calculated line parameters with those calculated on the basis of equivalent circular lines.

77-944

The Transient Characteristics of Fluid Pipes Including Mechanical Load (Third Report: Experimental Analysis on Effects of Valve Closure)

E. Kojima

Dept. of Engrg., Kanagawa Univ., Yokohama, Japan, Bull. JSME, 19 (136), pp 1182-1189 (Oct 1976) 14 figs, 5 refs

Key Words: Piping systems, Fluid-induced excitation

This paper deals with transient characteristics of a valve-controlled hydraulic driving system consisting of actuator, fluid pipe and operation valve, especially with pressure surge generation in a pipe due to sudden valve closure. Pressure variation was measured under various system and operating parameters, such as valve closure time, valve closure process, valve orifice shape, ets., and compared with theoretical results of the preceding paper.

77-945

The Transient Characteristics of Fluid Pipes Including Mechanical Load (Second Report: Theoretical Analysis on Effects of Valve Closure)

E. Kojima

Dept. of Engrg., Kanagawa Univ., Yokohama, Japan, Bull. JSME, 19 (136), pp 1172-1181 (Oct 1976) 11 figs, 8 refs

Key Words: Piping systems, Fluid-induced excitation

This paper presents a theoretical study of the transient characteristics for a system consisting of hydraulic actuator, fluid pipe and operation valve. In this analysis fluid motion in pipe is treated as a two-dimensional unsteady laminar flow, and non-linear characteristics of the flow through a valve is considered exactly.

77-946

Techniques for Controlling Piping Vibration and Failures

J.C. Wachel and C.L. Bates

Southwest Research Institute, San Antonio, TX, ASME Paper No. 76-Pet-18

Key Words: Piping systems, Vibration control

This paper describes techniques whereby piping vibration and failure problems can be controlled. Using the procedures outlined, the mechanical natural frequency of any straight piping span or piping bend can be calculated. The pulsation generation and coupling mechanisms that cause piping vibrations are described, and methods for predicting their frequencies are given.

77-947

The Dynamic Characteristics of an Electro-Hydraulic Servovalve

D.J. Martin and C.R. Burrows

The General Electric Co., Ltd., Hirst Res. Centre, Wembley, England, J. Dyn. Syst. Meas. and Control, Trans. ASME, 98 (4), pp 395-406 (Dec 1976) 7 figs

Key Words: Servomechanisms, Valves

The frequency responses of an experimental electro-hydraulic position control system and a simulation of the system are compared. Three different valve models are used in the simulation in an attempt to highlight the important parameters of an electro-hydraulic servovalve.

77-948

Aerodynamic Noise in Manifold Systems

D.A. Warden

Fisher Controls Co., Marshalltown, IA 50158, ISA Transactions, 15 (4), pp 354-358 (1976) 6 figs

Key Words: Manifolds, Noise generation

Many fluid transmission systems include a manifold fed by several lines with independent noise sources. This manifold often adds to and amplifies the noise radiated to the surroundings by the system. General observations are made on the nature and significance of this effect.

PLATES AND SHELLS

(Also see Nos. 866, 874, 898, 914, 929, 938)

77-949

Free Vibrations of Laminated Composite Elliptic Plates

C.M. Andersen and A.K. Noor

College of William and Mary, Williamsburg, VA., In: NASA, Langley Res. Ctr. Advan. in Eng. Sci., Vol 2, 1976, pp 425-438 (see N77-10265 01-31) N77-10270

Key Words: Plates, Laminates, Free vibration

Free vibrations of laminated anisotropic elliptic plates with clamped edges are studied. The analytical formulation is based on a Mindlin-Reissner type plate theory with the effects of transverse shear deformation, rotary inertia, and bending-extensional coupling included. The frequencies and mode shapes are obtained by using the Rayleigh-Ritz technique in conjunction with Hamilton's principle. A computerized symbolic integration approach is used to develop analytic expressions for the stiffness and mass coefficients and is shown to be particularly useful in evaluating the derivatives of the eigenvalues with respect to certain geometric and material parameters. Numerical results are presented for the case of angle-ply composite plates with skew-symmetric lamination.

77-950

Transient Response of Laminated Composite Plates Subjected to Transverse Dynamic Loading

J.M. Whitney and C. Sun

Air Force Materials Lab., Wright-Patterson AFB, OH 45433, J. Acoust. Soc. Amer., 61 (1), pp 101-104 (Jan 1977) 9 figs, 5 refs

Key Words: Plates, Composite structures, Laminates

Transient solutions are presented for an infinitely long, simply supported composite plate subjected to either a uniform or line concentrated dynamic pressure at the upper surface of the plate. A rectangular pulse, triangular pulse, and sinusoidal pulse are considered and dynamic load factors determined for maximum values of deflection, bending stress, and interlaminar shear stress as a function of pulse dwell time. Effect of pulse shape on deflection and stresses is also considered. These numerical results are obtained for graphite/epoxy symmetric angle-ply laminates.

77-951

Impact Loaded Plates. Calculation of Their Deflection and Bending Moments by the Bar Grate Method and Experimental Verification

W. Krings, V. Truppat and H. Waller

an der Ruhr-Universität Bochum, VDI Zeitschrift, 118 (24), pp 1189-1194 (Dec 1976) 12 figs, 4 refs (In German)

Key Words: Shock response, Plates

A numerical method for the calculation of dynamic behavior of plates is described. The method is based on a bar grate model of a plate; and, the impact, presented as a load function, enables the determination of plate deflection and section size. The bar grate method is particularly suitable for calculation of impact loads; because, even the complicated boundary conditions may be included in the calculation. The measurement and calculated results for bending were in very close agreement; for deflection the agreement was sufficient.

77-952

Numerical-Perturbation Technique for the Transverse Vibrations of Highly Prestressed Plates

A.H. Nayfeh and M.P. Kamat

Dept. of Engrg. Science and Mechanics, Virginia Polytechnic Inst. and State Univ., Blacksburg, VA 24061, J. Acoust. Soc. Amer., 61 (1), pp 95-100 (Jan 1977) 6 refs

Key Words: Plates, Flexural vibration, Perturbation theory

Under the usual assumptions of small strains with moderately large rotations, the problem of the transverse vibrations of highly prestressed, nonuniform annular plates is reduced to the solution of the differential equation governing the transverse vibration of the corresponding prestressed membrane subject to modified boundary conditions that account for the effects of bending. The methods of matched asymptotic and/or composite expansions are used to determine these modified boundary conditions. The agreement of the results of both methods with known exact solutions for simple geometries demonstrates the efficiency of this technique when compared with other well-known numerical techniques.

77-953

Higher-Order Effects of Initial Deformation on the Vibrations of Crystal Plates

X. Markenscoff

Virginia Polytechnic Inst. and State Univ., Blacksburg, VA, In: NASA, Langley Res. Ctr. Advan. in Eng. Sci., Vol. 1, 1976, pp 301-308 (see N77-10230 01-31)
N77-10259

Key Words: Plates, Initial deformation effects, Vibration response

A system of plate equations for the thickness-shear and flexural vibrations superposed on large initial deflection due to bending is derived. In the stress-strain relations, the terms associated with the fourth-order elastic stiffness coefficients are retained. An *explicit formula* for the change in the fundamental cut-off thickness shear frequency is obtained, and the effects of the terms associated with the fourth-order constants appear to be significant for large gradients of the rotation angles.

77-954

Wave Excited Oscillations of a Gravity Platform

C.L. Kirk and R. Nataraja

Cranfield Inst. of Technology, Cranfield, Bedford, England, ASME Paper No. 76-Pet-33

Key Words: Plates, Floating structures, Modal analysis

The dynamic response of a Condeep platform under transient linear wave excitation is determined by means of the modal method of analysis. The equations of motion for the fundamental torsional and bending modes are derived and solved by numerical step-by-step integration.

77-955

Natural Frequencies of Elastically Supported Orthotropic Rectangular Plates

E.B. Magrab

Institute for Basic Standards, National Bureau of Standards, Washington D.C. 20234, J. Acoust. Soc. Amer., 61 (1), pp 79-83 (Jan 1977) 4 figs, 13 refs

Key Words: Rectangular plates, Orthotropism, Natural frequencies, Rotary inertia effects, Transverse shear deformation effects, Boundary condition effects

An expression is derived from which the natural frequencies of a rectangular orthotropic plate, under any combination of simply supported, elastically supported, or clamped boundary conditions, can be obtained. The Mindlin-Timoshenko theory, which includes the effects of transverse shear and rotary inertia, is used to describe the plate motion. The solution is obtained with a previously developed extension of the Galerkin technique. Comparison of results with the limited results of previous investigations is very good. New results are presented for the fundamental frequencies of rectangular and square plates for boundary conditions on all four edges that vary continuously from simply supported to clamped, and for various combinations of length-to-thickness ratios and material constants. Additional results are presented for orthotropic plates simply supported and clamped on all four edges.

77-956

Vibration of an Acoustic Radiation from a Panel Excited by Adverse Pressure Gradient Flow

Y.M. Chang and P. Leehey

Acoustics and Vibration Lab., Massachusetts Inst. of Tech., Cambridge, MA, Rept. No. A/V-70208-12, 36 pp (May 1976)

AD-A032 070/5GA

Key Words: *Plates, Fluid-induced excitation, Vibration frequencies, Elastic waves*

Flow with uniform adverse pressure gradient was created over the surface of a steel test plate. Vibration velocity levels on the plate and acoustic radiation from the plate were measured and compared with theoretical estimates. The measured vibration levels agree with theory for frequencies above those for hydrodynamic coincidence.

77-957

Free Vibration Analysis of Cantilever Plates by the Method of Superposition

D.J. Gorman

Dept. of Mechanical Engrg., Univ. of Ottawa, Ottawa, Canada, J. Sound Vib., 49 (4), pp 453-467 (1976) 9 figs, 6 refs

Key Words: Free vibration, Cantilever plates, Method of superposition

The method of superposition is employed to analyze the first five symmetric and antisymmetric free vibration modes of a cantilever plate for a wide range of aspect ratios. It is shown that this method provides a simple, straightforward and highly accurate means of solution for this family of problems. Convergence to exact values is shown to be remarkably rapid. The first two symmetric and antisymmetric modal shapes for a square plate are accurately described by means of contour line drawings. The numerous advantages of this method over previously used methods are discussed.

77-958

Transient Response of a Cantilevered Plate to Impact Using Holographic Interferometry and Finite Element Techniques

J.C. MacBain

Air Force Aero Propulsion Lab., Wright-Patterson AFB, OH, Rept. No. AFAPL-TR-76-56, 71 pp (Aug 1976)

AD-A031 609/1GA

Key Words: Cantilever plates, Impact response (mechanical), Holographic techniques, Finite element techniques, Mathematical models, Computer programs

This report covers work carried out at AFAPL's Turbo Structures Research Laboratory (TSRL) on the transient structural response of an isotropic cantilevered plate subjected to normal impact by a ballistic pendulum. The program was a combined experimental/analytical effort. The experimental portion utilized a pulsed ruby laser to obtain holographic interferograms of the plate's deformation following impact. The analytical portion of the work consisted of mathematically modelling the plate using finite element techniques and studying the model's response to impact using the general purpose finite element program, NASTRAN.

77-959

Dynamic Inelastic Response of Thick Shells Using Endochronic Theory and the Method of Nearcharacteristics

H. Lin
Argonne National Lab., Argonne, IL., In: NASA, Langley Res. Ctr. Advances in Engrg. Sci., Vol. 2, 1976, pp 449-458 (see N77-10265 01-31)
Sponsored by ERDA
N77-10272

Key Words: Shells, Dynamic response

The endochronic theory of plasticity originated by Valanis was applied to study the axially symmetric motion of circular cylindrical thick shells subjected to an arbitrary pressure transient applied at its inner surface. The constitutive equations for the thick shells were obtained. The governing equations are solved by means of the nearcharacteristics method.

77-960

Application of One-Dimensional Finite Elements to Normal Mode Analysis of Ferroelectric Cylindrical Shells

H.A. Sabbagh and T.F. Krile
Naval Weapons Support Center, Crane, IN 47522, J. Acoust. Soc. Amer., 61 (1), pp 2-10 (Jan 1977) 9 figs, 4 refs

Key Words: Cylindrical shells, Modal analysis, Normal modes, Finite element techniques

System models, consisting of linear partial differential equations which have been put into variational form, can be solved using the finite element method. The construction and use of a special class of one-dimensional finite elements, consisting of multiple convolutions of the unit pulse function, is described in this paper.

77-961

Effect of Bending on the Axisymmetric Vibrations of a Spheroidal Model of the Head

T.T. Lee
Ph.D. Thesis, Columbia Univ., 53 pp, 1976
UM 76-29,599

Key Words: Spherical shells, Fluid-filled containers, Heads (anatomy), Free vibration, Mathematical models

The head is modelled as an elastic prolate spheroidal shell filled with an inviscid, compressible fluid. Bending effects are included, and the free vibration frequency spectrum obtained is compared with that of an earlier spheroidal model using extensional (membrane) shell theory and with a spherical model including bending. The differences between the present results and those reported previously are significant.

77-962

Finite Element Analysis of a Seismically Excited Cylindrical Storage Tank, Ground Supported and Partially Filled with Liquid

S.H. Shaaban and W.A. Nash
Dept. of Civil Engrg., Massachusetts Univ., Amherst, MA., Rept. No. NSF/RA-760261, 116 pp (July 1976)
PB-258 506/5GA

Key Words: Storage tanks, Fluid-filled containers, Seismic response, Finite element technique, Shells

The structure under consideration is an elastic cylindrical liquid storage tank attached to a rigid base slab. The tank is filled to an arbitrary depth with an inviscid, incompressible liquid. A finite element analysis is presented for the free vibrations of the coupled system, permitting determination of natural frequencies and associated mode shapes. The response of the partially-filled tank to artificial earthquake excitation is also determined through use of finite elements. Examples, together with program listing, are offered.

77-963

Fluid-Loading Influence Coefficients for a Finite Cylindrical Shell

B.E. Sandman
Naval Underwater Systems Ctr., Newport, RI 02840, J. Acoust. Soc. Amer., 60 (6), pp 1256-1264 (Dec 1976) 8 figs, 19 refs

Key Words: Cylindrical shells, Fluid-induced excitation

The fluid radiation loading exhibited by a finite cylindrical shell with rigid immovable end plates is investigated by implementing the methods of Fourier integral transforms. The solution is obtained for a generalized velocity distribution on the cylindrical surface. The numerical solution for a radially pulsating surface is utilized as the basis for comparison to previous investigations. In addition, the radiation loading displayed by the low-order modes of a simply supported shell section is obtained, and the relative effect of the imposed condition of an infinite cylindrical baffle embodying the shell is ascertained.

77-964

Hyperbolic Cooling Towers. Part I

P. Rogers
California State Univ., Los Angeles, CA, Mecaniqué
Appliquée, 21 (2), pp 183-199 (1976) 11 figs, 21 refs

Key Words: Cooling towers, Hyperbolic parabolic shells, Wind induced excitation, Seismic excitation

A study is presented that includes both the theoretical-practical aspect of the design and the construction of hyperbolic cooling towers. As these towers reach immense sizes, more substantial wind and earthquake assumptions are required, and the analysis of the shells and other component parts must be based on advanced methods, with emphasis on dynamic response analysis against seismic forces.

77-965

Hyperbolic Cooling Towers. Part II

P. Rogers
California State Univ., Los Angeles, CA, Mecaniqué
Appliquée, 21 (3), pp 371-387 (1976) 21 figs

Key Words: Cooling towers, Hyperbolic parabolic shells, Wind induced excitation, Seismic excitation

A study is presented that includes both the theoretical and practical aspect of the design and construction of hyperbolic cooling towers. As these towers reach immense sizes, more substantial wind and earthquake assumptions are required, and the analysis of the shells and other component parts must be based on advanced methods, with emphasis on dynamic response analysis against seismic forces.

77-966

Dynamic Response of an Axisymmetric Lined Cylindrical Cavity of Finite Length in a Bilinear Material

A.T. Matthews and H.H. Bleich
Weidlinger Associates, NY, Rept. No. DNA-3997F, 66 pp (Apr 1976)
AD-A031 760/2GA

Key Words: Underground structures, Cylindrical shells, Interaction: soil-structure

A pressure pulse ρ (theta) is applied to the interior of an axisymmetric, lined cylindrical cavity of finite length, embedded in a bilinear type dissipative material. The cavity lining is a structure composed of a thin, axisymmetric cylindrical shell with bending stiffness, welded to a flat circular plate at the bottom. The response of the dissipative material is found from a pseudo-characteristic scheme combined with a fractional step method (and given the name PC scheme). Results are presented and compared with results from a finite element scheme applied to the same configuration. These comparisons illustrate the use of this PC scheme code as a check solution for a large computer code such as the finite element one.

77-967

Influence of Asymmetric Imperfections on the Vibrations of Axially Compressed Cylindrical Shells

A. Rosen and J. Singer
Dept. of Aeron. Engrg., Technion -- Israel Institute of Technology, Haifa, Israel, Israel J. Tech., 14 (1-2), pp 23-26 (1976) 5 figs, 3 refs
Sponsored by the Air Force Office of Scientific Res.

Key Words: Cylindrical shells, Initial deformation effects, Geometric information effects

The influence of two types of asymmetric initial geometrical imperfections on the vibrations and buckling of axially loaded isotropic cylindrical shells is studied. The analysis is based on a solution of the Kármán-Donnell nonlinear shell equations by the Galerkin method. The results indicate that the influence of the initial geometrical imperfections depends strongly on the mode of the initial imperfection and its relation to the mode of vibration. The imperfections may lower or raise the frequency. The buckling loads also depend strongly on the relation between the corresponding modes.

SYSTEMS

ABSORBER

(Also see Nos. 871, 872)

77-968

The Response of a System When Modified by the Attachment of an Additional Sub-System

R.G. Jacquot
Dept. of Elect. Engrg., Univ. of Wyoming, Laramie, WY 82071, J. Sound Vib., 49 (3), pp 345-351 (1976) 7 figs, 14 refs

Key Words: *Damped structures, Optimization,* Cantilever beams, Dynamic vibration absorption (equipment)

A technique is developed by which the effect of the attachment of a sub-system on frequency domain and time domain responses may be evaluated. This involves working with driving point and transfer functions in the frequency domain. Transition to the time domain is accomplished by use of the Fast Fourier Transform algorithm. The technique developed is applied to find the optimal damping for a dynamic vibration absorber attached to the tip of a forced elastic cantilever beam.

NOISE REDUCTION

(Also see Nos. 877, 934, 975, 988, 1001, 1002, 1015, 1035)

77-969

Acoustical Analysis, Testing, and Design of Flow-Reversing Muffler Chambers

C.J. Young and M.J. Crocker
Engrg. Research and Development Div., Engrg. Dept., E. I. DuPont de Nemours and Co., Wilmington, DE 19898, J. Acoust. Soc. Amer., 60 (5), pp 1111-1118 (Nov 1976) 21 figs, 4 refs

Key Words: Mufflers, Sound transmission loss, Mathematical models

The transmission loss characteristics of flow-reversing muffler chambers were predicted by a numerical approach based on the finite element method. The theoretical model developed is described in this paper; its validity is established experimentally with a number of different chambers. The standing wave method was used to measure transmission loss and measurements were conducted with and without steady air flow.

77-970

Diffraction of Sound by Nearly Rigid Barriers

W.J. Hadden, Jr. and A.D. Pierce
Georgia Inst. of Tech., Atlanta, GA, In: NASA Langley Res. Ctr. Advances in Engrg. Sci., Vol. 3 1976, pp 1009-1018 (See N77-10305 01-31) N77-10322

Key Words: Acoustic scattering, Noise barriers

The diffraction of sound by barriers with surfaces of large, *but finite, acoustic impedance was analyzed.* Idealized source-barrier-receiver configurations in which the barriers may be considered as semi-infinite wedges are discussed. Particular attention is given to situations in which the source and receiver are at large distances from the tip of the wedge. The expression for the acoustic pressure in this limiting case is compared with the results of Pierce's analysis of diffraction by a rigid wedge. An expression for the insertion loss of a finite impedance barrier is compared with insertion loss formulas which are used extensively in selecting or designing barriers for noise control.

77-971

An Example of In-Plant Noise Reduction with an Acoustical Barrier

J.B. Moreland and R.F. Minto
Westinghouse Electric Corp., Pittsburgh, PA 15235, Appl. Acoust., 9 (3), pp 205-214 (July 1976) 7 figs, 3 refs

Key Words: Industrial facilities, Machinery noise, Noise barriers

Existing theories for predicting the distribution of sound intensity in rooms and the performance of acoustical barriers are applied in designing a barrier to reduce noise in an industrial environment. The combination of the theories is found to predict the sound pressure level in the barrier shadow zone reasonably well.

77-972

The Technical Feasibility of Noise Control in Industry

Bolt Beranek and Newman Inc., Cambridge, MA Rept. No. OSHA-EFS-76-800, 329 pp (Aug 1976) PB-259 792/0GA

Key Words: Noise reduction, Machinery noise

Machines are listed under standard industrial codes to show industries which use each machine. Machines are then assigned to 3 categories of noise problems: no noise problem, noise problem identified, and noise problem unassigned. The unassigned at present contains most of the machines listed. Machines are also categorized as one of 5 types: process, not specific (a basic manufacturing process), heating and ventilating, material handling, and packaging. Where noise problems are identified, possible engineering controls are listed. Engineering controls considered include replacement of noise producing devices, treatments for large radiating surfaces, mufflers, enclosures, barriers, lagging, vibration isolation, personnel acoustic booths, and room acoustic treatment. Portions of this document are not fully legible.

AIRCRAFT

(Also see Nos. 858, 880, 886, 887, 905, 910)

77-973

An Analytical and Experimental Investigation of the Hovering Dynamics of the Aerocrane Hybrid Heavy Lift Vehicle

W.F. Putman and H.C. Curtiss

All American Engrg. Co., Wilmington, DE, Rept. No. OS-137, 96 pp (June 1976)

AD-A031 443/5GA

Key Words: Cargo aircraft, Hovercraft

The results of an analytical and experimental investigation of the hovering dynamics of an AEROCRANE hybrid heavy lift vehicle are discussed and compared. Analytical representations of the hovering aerodynamics and equations of motion are developed and discussed. The experimental program, including flight test of a 0.107 scale Froude model, is discussed. Piloted and unpiloted analog computer simulations of the hovering dynamics are compared with flight test results. The influence of feed-back stabilization is investigated and recommendations made for a stabilization system to improve the hovering operation by a remote pilot with limited motion cues.

77-974

Inflight Flutter Identification of the MRCA

D.K. Potter and A. Lotze

British Aircraft Corp., Warton, England, In: AGARD Structural Identification on the Ground and in Flight Including Command and Stability Augmentation System Interaction, June 1976, pp 25-39 (avail: N76-29656 20-39)

N76-29659

Key Words: Aircraft, Flutter

Flutter investigations were performed prior to flight testing and during flight flutter testing of the multi-role combat aircraft (MRCA). Because the aircraft is equipped with fast responding power control systems which could produce undesirable structural motion, flutter investigations had to be accomplished with consideration of the command and stability augmentation system (CSAS). Analysis and test results for structural mode coupling with the CSAS are demonstrated for the aircraft on ground which proved to be the condition for the lowest stability margin. It was shown that there is practically no influence of CSAS on flutter behavior. The flutter speed with the lowest flutter margin was predicted for an antisymmetrical taileron mode which is modified by fuselage influences. The coupling mechanism of this mode was investigated and the effect of apex balance weight on the taileron inboard leading edge was demonstrated.

77-975

Noise Technology for Future Aircraft Power Plants

J.D. Kester and A.A. Peracchio

Pratt & Whitney Aircraft, East Hartford, CT., Mech. Engr., 99 (1), pp 40-47 (Jan 1977), 15 figs, 13 refs

Key Words: Aircraft engines, Noise reduction, Ducts, Acoustic linings

Noise technology requirements are surveyed for several future aircraft power plants. Applications range from growth versions of current production engines to future advanced supersonic transport applications. Although a variety of complex and challenging noise-suppression problems are identified, a sampling of several basic problems common to a variety of engine designs are discussed in detail. The prediction of fan noise generation and propagation in treated ducts, the use of mixer nozzles to reduce jet exhaust noise, and the prediction and absorption of combustion noise.

77-976

The Response of a Vibrating Structure as a Function of Structural Parameters -- Application and Experiment

G.T.S. Done, A.D. Hughes, and J. Webby
Dept. of Mech. Engrg., Univ. of Edinburgh, Edinburgh EH9 3JL, Scotland, J. Sound Vib., 49 (2), pp 149-159 (Nov 22, 1976) 6 figs, 6 refs

Key Words: Aircraft seats, Mathematical models

The theory used for examining the response of a structure subject to an external oscillatory force when certain structural parameters are changed is applied to the mathematical model of a pilot's seat structure. The results are discussed with a view to assessing those parts of the structure that are most effective in minimizing the response. Also, a laboratory experiment is described in which the basic circular response locus is verified, and doubts about the assumptions made in mathematical modeling resolved.

77-977

Statistical Analysis of the Vibration Response of External Aircraft Stores

G. Hadan and R. Eshel
Dept. of Mech. Engrg., Technion -- Israel Institute of Technology, Haifa, Israel, Israel J. Tech., 14 (1-2), pp 86-93 (1976) 12 figs, 7 refs

Key Words: Wing stores, Vibration response, Statistical analysis

Vibrations in external stores of the Phantom F4E were measured and analyzed for various flight conditions. Statistical relationships between the vibration levels and dynamic pressure, Mach number, store structure, etc., are observed and explained.

BIOENGINEERING

(Also see No. 961)

77-978

Biodynamics of Deformable Human Body Motion

A.M. Strauss and R.L. Huston
Cincinnati Univ., Cincinnati, OH, In: NASA, Langley Res. Ctr. Advances in Engrg. Sci., Vol. 1 1976, pp 309-318 (See N77-10230 01-31)
N77-10260

Key Words: Biomechanics, Test models

The objective is to construct a framework wherein the various models of human biomaterials fit in order to describe the biodynamic response of the human body. The behavior of the human body in various situations, from low frequency, low amplitude vibrations to impact loadings in automobile and aircraft crashes, is very complicated with respect to all aspects of the problem: materials, geometry and dynamics. The materials problem is the primary concern, but the materials problem is intimately connected with geometry and dynamics.

77-979

Oblique Impact on a Head-Helmet System

B.A. Simpson, W. Goldsmith, and J.L. Sackman
Univ. of California, Berkeley, CA 94700, Intl. J. Mech. Sci., 18, pp 337-340 (July/Aug 1976), 4 figs, 7 refs

Key Words: Helmets, Impact tests

The propagation of waves produced by pendulum impact both radially and obliquely on two different models simulating a head-helmet system was studied experimentally by means of strain gages attached to the surfaces of the components representing the skull and the helmet and by tourmaline crystals embedded in a gel representing the brain. The first model consisted of a set of nested aluminum and styrofoam full or partial spherical shells, while the second comprised an actual cadaver skull held in an army helmet by means of a polyurethane foam. The results were compared with theoretical and experimental information previously developed.

BRIDGES

(Also see No. 864)

77-980

Self-Excited Aeroelastic Instability of a Bridge Deck

J.G. Beliveau
Universite de Sherbrooke, Sherbrooke, Quebec, Canada, ASME Paper No. 76-WA/FE-25

Key Words: Suspension bridges, Flutter, Critical speed

The self-excited loads due to the response of a suspension bridge are critical to the stability of the motion. Although buffeting loads due to the random components of the wind enter into the mean square of the response, the response is dominated by the inherent instability of the system near the critical velocity. In this study of a particular deck, experimental information in the form of quasisteady aerodynamic coefficients on an actual bridge deck is used to determine the flutter speed.

77-981
On Vortex Shedding Excitations of Cable-Stayed Bridge of Closed Polygonal Cross Sections
N. Shiraishi and M. Matsumoto
Dept. of Civil Engrg., Kyoto Univ., Kyoto, Japan,
Mem. Fac. Engr., Kyoto Univ., 38 (2), pp 37-54 (Apr 1976) 9 figs, 30 refs

Key Words: Suspension bridges, Vortex shedding, Fluid-induced excitation

This paper presents a few fundamental instability characteristics of vortex shedding excitations for closed cross sections of bridge structures by means of wind tunnel and water flume tests. Experimental results indicate that the hexagonal form of a cross section with spoilers and flaps is a satisfactory stable cross section for aeolian oscillations.

BUILDING
(Also see Nos. 885, 964)

77-982
Evaluation of Seismic Safety of Buildings
Report No. 8. Inelastic Response Spectrum Design Procedures for Steel Frames
R.W. Haviland, J.M. Biggs, and S.A. Anagnostopoulos
Dept. of Civil Engrg., Massachusetts Inst. of Tech., Cambridge, MA,
Rept. No. R76-40, 146 pp (Sept 1976) (Also see Rept. No. PB-252 898, Jan 1976)
PB-258 559/4GA

Key Words: Seismic design, Buildings

Reported is an evaluation of aseismic design procedures based upon inelastic response spectra. Steel frames of different heights are designed for a desired level of yielding using elastic modal analysis. Responses in terms of maximum ductility ratios are computed for simulated ground motions derived from the design spectrum. Both shear beam models and point hinge models are utilized and compared. Results are given in terms of maximum local and story ductility ratios as compared with the design values. The effect of gravity loads on the computed response and the effect of including such loads in the design procedure are investigated.

77-983
The Seismic Behavior of Critical Regions of Reinforced Concrete Components as Influenced by Moment, Shear, and Axial Force
M.B. Atalay and J. Penzien
Earthquake Engrg. Research Ctr., California Univ., Berkeley, CA,
Rept. No. EERC-75-19, 250 pp (Dec 1975)
PB-258 842/4GA

Key Words: Buildings, Earthquake response

To determine the characteristics and modes of failure of columns due to earthquake induced degradations in stiffness, strength, and energy absorption, a series of twelve members simulating a column between inflection points above and below a floor level were designed and tested dynamically. The variable parameters introduced were magnitude of applied axial load chosen to represent lower, intermediate, and upper story columns, lateral reinforcement percentage chosen to study the influence of confinement on ductility, and history of controlled lateral displacement chosen to determine the effects of rate and sequence loading.

77-984
Some Dynamic Problems of Rotating Windmill Systems
J. Dugundji
Massachusetts Inst. of Tech., Cambridge,
In: NASA, Langley Res. Ctr. Advances in Engrg. Sci., Vol. 2 1976, pp 439-447 (See N77-10265 01-31)
N77-10271

Key Words: Windmills, Towers

The basic whirl stability of a rotating windmill on a flexible tower is reviewed. Effects of unbalance, gravity force, gyroscopic moments, and aerodynamics are discussed. Some experimental results on a small model windmill are given.

CONSTRUCTION

(Also see No. 998)

77-985

Simple Valving Creates Nearly-Vibrationless Jackhammer

B.J. Hogan

Design Ideas, pp 46-47

Key Words: Pneumatic equipment, Construction equipment, Hammers, Vibration control

The design of a system of valves to minimize jackhammer vibration is described.

EARTH

77-986

Evaluation of In Situ Shear Wave Velocity Measurement Techniques

A. Viksne

Engrg. and Research Ctr., Bureau of Reclamation, Denver, CO 80225, Rept. No. REC-ERC-76-6, 40 pp (Apr 1976)

Key Words: Dams, Secondary waves, Measurement techniques

Several geophysical methods used by the Bureau of Reclamation to obtain the in situ shear wave velocity of earth embankments were evaluated. In situ low-strain shear wave velocity determinations have been performed on a number of existing zoned earthfill dams and an earth dam under construction. Test methods and results of the various in situ measurements are compared and evaluated.

ELECTRICAL

77-987

Influence of Skin Effect on Steady-State Operation and Forced Oscillations of Asynchronous Machines

A. Meyer

Baden, Switzerland, Brown Boveri Rev., 63, pp 500-507 (Aug 1976) 11 figs, 8 refs

Key Words: Electrical machines, Rotors, Vibration damping, Dynamic stability

As unit ratios increase it becomes increasingly difficult, if not impossible, to carry out heating measurements on asynchronous machines by conventional means in the test laboratory. For this reason new heat-run methods are becoming more and more important. On the basis of the system equations this article examines steady-state operation of three-phase squirrel-cage machines with skin effect in the rotor. The article also studies the influence of skin effect on dynamic stability and elasticity and damping in asynchronous machines when a harmonic oscillating torque is applied to the shaft in addition to an opposing torque independent of time. The forced oscillations of the rotor and the resulting mains power fluctuations are calculated for an asynchronous squirrel-cage machine of about 3 MW.

77-988

Aerodynamic Noise in Medium-Sized Asynchronous Motors

B. Ploner

Baden, Switzerland, Brown Boveri Rev., 63, pp 493-499 (Aug 1976) 11 figs, 6 refs

Key Words: Electrical machines, Rotors (machine elements), Noise reduction, Acoustic absorption

In a medium-sized asynchronous machine, the importance of the aerodynamic noise component, against that produced by electro-magnetic forces, is continually increasing, because to increase the power-weight ratio the air-cooling must be intensified. Where, for example, the fan-running noise is only insufficiently influenced by measures taken at source, interesting solutions often present themselves through reduction of the propagation of sound by means of sound-proofing and sound-absorbent elements. After a short introduction to the basic principles of such component parts, their practical use is described by means of some examples.

FOUNDATIONS

77-989

Design of Framed Foundation Subjected to Vibration

C.H. Nam, N.V. Campomanes, and Y.K. Kim
The Ralph M. Parsons Co., Pasadena, CA 91124,
Intl. J. Computers and Structures, 7 (1), pp 103-107 (Feb 1977) 8 figs, 8 refs

Key Words: Foundations, Equipment mounts, Reinforced concrete, Natural frequencies, Beams

The potential effect of repeated loads on the stiffness and the natural frequencies of framed foundations which support vibrating equipment such as compressors or turbine generators are described. The reason for the reduction of stiffness of concrete under repeated loads is explained. A two-dimensional element and a bar element are utilized to idealize reinforced concrete structures. Principal stresses are checked to determine the status of cracking of concrete. If the principal stress is greater than the tensile capacity of concrete, a stress crack is assumed to develop in the perpendicular direction to the corresponding stress. The element stiffness is modified to account for the crack. The modulus of elasticity of concrete is modified under repeated loads according to the stress state. The modulus of elasticity of concrete is assumed to be constant in compression, but varying in tension. Example beams are analyzed to study the effect of the reduction of modulus in tension zone.

HELICOPTERS

(Also see No. 869)

77-990

Dynamic Characteristics of Rotor Blades

V.R. Murthy
Dept. of Mech. Engrg. and Mechanics, Old Dominion Univ., Norfolk, VA 23508,
J. Sound Vib., 49 (4), pp 483-500 (Dec 22, 1976) 29 refs

Key Words: Rotary wings, Natural frequencies, Mode shapes

The transmission matrix method is used to determine the dynamic characteristics (natural frequencies and the associated mode shapes) of rotor blades. The problems treated are combined flapwise bending, chordwise bending, and torsion of twisted non-uniform blades and its special subcases. The orthogonality relations that exist between the natural modes are derived. The method is appealing because of its simplicity for programming for digital computer calculations and also the inputs are generated very easily since they are merely the coefficients of the differential equations of motion.

77-991

Dynamics of a Slung Load

L. Feaster, C. Poli, and R. Kirchhoff
Univ. of Massachusetts, Amherst, MA,
J. Aircraft, 14 (2), pp 115-121 (Feb 1977) 12 figs, 13 refs

Key Words: Cargo aircraft, Helicopters, Shipping containers, Suspended structures

The yaw damping coefficient of an 8X8X20 ft cargo container is experimentally determined by forced oscillation tests. The result is then used in linearized small-perturbation stability analyses of a slung load, considering both single-cable and two-cable tandem suspension systems. The effect of attaching stabilizing fins to the container and of incorporating a rotating wheel for stability augmentation is also theoretically investigated.

77-992

Random Vibration Peaks in Rotorcraft and the Effects of Nonuniform Gusts

G.H. Gaonkar
Southern Illinois Univ., Edwardsville, IL 62025,
J. Aircraft, 14 (1), pp 68-76 (Jan 1977) 9 figs, 22 refs

Key Words: Helicopters, Random vibration, Rotor blades, Wind-induced excitation

The analysis of random blade vibrations is extended to include the average number of peaks above arbitrary thresholds, and to the case of both longitudinally and laterally nonuniform or completely nonuniform vertical turbulence over the rotor disk. This extended analysis provides a means of assessing the validity of uniform and partially nonuniform (nonuniform only in the longitudinal direction) approximate turbulence models used earlier. Further, it exactly identifies threshold ranges above which peak distribution functions (PDF) over one rotor revolution can be approximated by the statistics of threshold upcrossings.

77-993

Elementary Applications of a Rotorcraft Dynamic Stability Analysis

W. Johnson
NASA, Ames Research Ctr., Moffett Field, CA,
Rept. No. NASA-TM-X-73161; A-6717, 24 pp (June 1976)
N76-33129

Key Words: Rotors, Helicopters, Dynamic stability

A number of applications of a rotorcraft aeroelastic analysis are presented to verify that the analysis encompasses the classical solutions of rotor dynamics, and to examine the influence of certain features of the model. Results are given for the following topics: flapping frequency response to pitch control; forward flight flapping stability; pitch/flap flutter and divergence; ground resonance instability; and the flight dynamics of several representative helicopters.

77-994

Noise Certification Considerations for Helicopters Based on Laboratory Investigations

Man-Acoustics and Noise Inc., Seattle, WA, Rept. No. MAN-1014, FAA-RD-76-116, 108 pp (July 1976) (Also see Rept. No. AD-A018 036, Nov 1975)
AD-A032 028/3GA

Key Words: Helicopter noise, Human factors engineering

This is the second part of a program concerning noise certification for V/STOL and helicopter aircraft. Aspects considered were: an engineering calculation procedure which validly and reliably reflects annoyance to helicopter operations; estimates of noise exposure levels which could be compatible with human activities in areas surrounding heliports; noise exposure modeling for helicopter noise; certification measurement approaches for helicopter noise certification. The basics of the program involved human response evaluations of conventional takeoff and landing (CTOL) aircraft noise, simulations of helicopter noise emphasizing 'slap' or pulsating noise effects, and recordings of a wide variety of helicopter operations.

HUMAN

(Also see Nos. 978, 1028)

77-995

Subjective Equivalence of Sinusoidal and Random Whole-Body Vibration

M.J. Griffin
Human Factors Res. Unit, Inst. of Sound and Vibration Research, Univ. of Southampton, Southampton S09 5NH, England, J. Acoust. Soc. Amer., 60 (5), pp 1140-1145 (Nov 1976) 4 figs, 12 refs

Key Words: Random excitation, Human response

An experiment conducted to compare the discomfort produced by whole-body sinusoidal vibration with that produced by one-third-, one-, and three-octave vibration spectra is described.

ISOLATION

77-996

Shock Spectrum of a Two-Degree-of-Freedom Nonlinear Vibratory System

R.M. Root and P.F. Cunniff
Center for Naval Analyses, Arlington, VA 22209, J. Acoust. Soc. Amer., 60 (6), pp 1314-1318 (Dec 1976) 5 figs, 21 refs

Key Words: Shock isolators, Foundations, Vibrating structures

The shock spectrum of the foundation or isolator mass of a two-degree-of-freedom vibratory system was studied. The system had a cubic hardening elastic nonlinearity in the foundation or isolator restoring force. The system was impulsively shocked, and analytical, experimental, and numerical methods used to determine the resulting shock spectrum. The system was studied theoretically in two ways. An analytic solution was developed using a perturbation expansion of the nonlinear equations of motion, combined with an analytic solution for the shock spectrum due to the motion. A numerical solution to the nonlinear equations of motion was developed and used to verify the range of validity of the solution developed from the perturbation expansion. An experimental study of a two-degree-of-freedom system with a cubic hardening spring was undertaken. The experimental results verified shifting of the frequencies of the peaks and introduction of additional peaks in the shock spectrum.

77-997

Studies on Realization of Optimum Vibration Isolators for Systems with Random Excitations

N. Fujiwara and Y. Murotsu
College of Engrg., Univ. of Osaka Prefecture, Sakai, Osaka, Bull. JSME, 19 (136), pp 1129-1134 (Oct 1976) 11 figs, 4 refs

Key Words: Vibration isolators, Random excitation, Optimization

This paper is devoted to realization of the optimum vibration isolators for systems subjected to stationary random excitations. A systematic procedure expressing the optimum vibration isolating forces as functions of the state variables is proposed and illustrated by examples of a single- and a two-degree-of-freedom systems.

MATERIAL HANDLING

77-998

Dynamic Response of a Portable Level-Lifting Crane during Torsion and Lifting

Fortschritt-Ber. der VDI Zeitschr., Series 13, No. 14 (1976) (summarized in VDI-Zt., 118 (22), p 1080 (Nov 1976) by Dr. M.A. Parameswaran) Avail: VDI Verlag GmbH, Postfach 1139, 4000 Düsseldorf 1

Key Words: Cranes, Torsional response, Mathematical models

The report describes an investigation of torsional and lifting motions of a portable lifting crane. The torque of the drive shaft, the oscillation of the suspended load and the vibration of the boom are the main quantities examined. The analytical model was tested and all the measurements were carried out on a crane located in the harbor of Madras, India.

MECHANICAL

(See No. 892)

METAL WORKING AND FORMING

77-999

The Identification of the Source of Machine Noises Contained Within a Multiple-Source Environment

R.A. Collacott

UK Mechanical Health Monitoring Group, Great Glen, Leicestershire, Great Britain, Appl. Acoust., 9 (3), pp 225-238 (July 1976) 11 figs, 3 refs

Key Words: Machinery noise, Noise source identification, Machine tools

This investigation explored the response of six machine tools in a workshop when operating individually (solo) and together (simultaneously). Studies are made of the respective time-domain waveforms, decibel-frequency spectra (sound signatures) and power spectral density-frequency spectra. A comparison has been made between the resulting power spectral density spectra when all machines are working together and the computer-summated spectra of the machines when running individually.

77-1000

Effect of a Variable Stiffness-Type Dynamic Damper on Machine Tools with Long Overhung Ram

K. Seto and N. Tominari

National Defense Academy, Yokosuka, Japan, Bull. JSME, 19 (137), pp 1270-1277 (Nov 1976) 21 figs, 12 refs

Key Words: Machine tools, Chatter, Tuned dampers

The paper reports the results of an examination into the performance of a variable stiffness-type dynamic damper which is intended to increase the cutting performance of machine tools with a long overhung ram. The spring stiffness of the damper is changed externally by adjusting the position of a damper mass. Investigation is made to establish a condition for tuning the damper to maximum efficiency when changes in the length of the ram take place in cutting operations.

PUMPS, TURBINES, FANS, COMPRESSORS

(Also see Nos. 895, 908, 989)

77-1001

Noise Reduction in Centrifugal Fans: A Literature Survey

W. Neise

Inst. of Sound and Vibration Res., Southampton Univ., Southampton, England, Rept. No. ISVR-TR-76, 47 pp (June 1975)

N77-10868

Key Words: Fans, Noise reduction, Reviews

In order to help the industrial engineer construct quieter fans, the research work done by various experimenters during the last fifteen years on noise reduction methods in centrifugal fans used in air conditioning and ventilating equipment is summarized. Only those investigations are included in which an attempt was made to reduce the generated sound by modifying the fan itself. Most of the work described was aimed at reducing the blade passage sound, but in some cases a decrease of the broad band noise was also obtained. Three different centrifugal impeller designs are dealt with: multivane impellers with forward curved blades, impellers with backward curved blades, and impellers with radial blades. The following methods are described: increasing the distance between impeller tip and cutoff, angle of inclination between impeller blades and cutoff edge, staggering of blades of double inlet blowers and double row impellers, and transition meshes at the leading and trailing edges of the rotor blades. Future research work on the main source regions of broadband noise is suggested.

77-1002
Effects of Long-Chord Acoustically Treated Stator Vanes on Fan Noise. 2: Effect of Acoustical Treatment
J.H. Dittmar, J.N. Scott, B.R. Leonard, and E.G. Stakolich
NASA, Lewis Research Ctr., Cleveland, OH., Rept. No. NASA-TN-D-8250, E-8736, 99 pp (Oct 1976)
N76-33206

Key Words: Fans, Noise reduction, Acoustic linings

A set of long chord stator vanes was designed to replace the vanes in an existing fan stage. The long chord stator vanes consisted of a turning section and axial extension pieces, all of which incorporated acoustic damping material. The long chord stator vanes were tested in two lengths, with the long version giving more noise reduction that the short, primarily because of the additional lining material.

77-1003
The Influence of Viscosity of the Working Fluid on the Sound Production of Centrifugal Fans
M. Bartenwerfer and R. Agnon
Inst. fuer Turbulenzforschung, Deutsche Forschungs- und Versuchsanstalt fuer Luft- und Raumfahrt, Berlin, West Germany, Rept. No. DLR-FB-76-30, 23 pp (May 18, 1976)
(In German)
N76-33955

Key Words: Fans, Sound generation

The influence of working fluid viscosity on the sound production of centrifugal fans was experimentally studied using a model fan with backward curved blades. The Reynolds number based on tip speed and diameter of the impeller varied over a range of Reynolds numbers by changing the air density at a constant temperature. Measurements of the harmonic and the random noise components are made at several points in the acoustic near field as well as in the far field.

77-1004
Experimental Study of the Influence of the Casing on the Noise Radiation of Centrifugal Fans
R. Agnon
Forschung im Ingenieurwesen, 42 (6), pp 187-200 (1976) 22 figs, 24 refs
(In German)

Key Words: Fans, Noise generation

The aerodynamic noise production of a centrifugal fan depends on the acoustic loading of the noise sources; i.e., the sound spectrum is determined not only by the fluid flow but also by acoustic resonance in the casing. In the present study this effect is investigated with respect to the blade passage noise. The radiated sound pressure of the harmonic noise is analytically described by a product of non-dimensional terms. One of these, called the frequency response function, contains the parameters relevant to the acoustic characteristics of the casing.

77-1005
The Inlet Vortex as a Source of Tonal Rotor Noise
L.J. Leggat
Ph.D. Thesis, The Univ. of British Columbia, Canada

Key Words: Fans, Noise generation

The thesis describes experimental and mathematical analyses of the noise resulting from the interaction of an axial flow fan with a concentrated inlet vortex. A comparison of results from the two methods reveals the physical phenomena on which the discrete tone noise depends.

77-1006
Damping of Fluid Vibrations in Hydraulic Oil Systems
D. Hoffmann
VDI Forsch.-Heft 575, Düsseldorf, VDI Verlag 1976, 48 pp, 107 figs (summary in VDI-Zeitschrift, 118 (22) (Nov 1976) by H.W. Hahnemann)
(In German)

Key Words: Vibration damping, Fluid drives

The author describes, in detail, vibration excitation in hydraulic oil systems. The modes of propagation of noise as well as its control are discussed.

77-1007
Tower and Rotor Blade Vibration Test Results for a 100-Kilowatt Wind Turbine
B.S. Linscott, W.R. Shapton and D. Brown
NASA, Lewis Research Center, Cleveland, OH , Rept. No. NASA-TM-X-3426, E-8751, 40 pp (Oct 1976)
N76-33628

Key Words: Wind turbines, Turbines, Turbine blades, Towers, Natural frequencies, Mode shapes, Computer programs

The predominant natural frequencies and mode shapes for the tower and the rotor blades of the ERDA-NASA 100-kW wind turbine were determined. The tests on the tower and the blades were conducted both before and after the rotor blades and the rotating machinery were installed on top of the tower. The tower and each blade were instrumented with an accelerometer and impacted by an instrumented mass. The tower and blade structure was analyzed by means of NASTRAN, and computed values agree with the test data.

77-1008

How to Avoid Field Problems with...Boiler Feed Pumps

E. Makay
Energy Research & Consultants Corp., Morrisville, PA., Hydrocarbon Processing, 55 (12), pp 79-84 (Dec 1976) 11 figs

Key Words: Pumps, Rotor-bearing systems, Failure analysis

Hydraulically induced forces that can influence rotor and bearing design requirements are described. Rotor stability and pump rotor critical speeds are changed.

77-1009

A Compressor Manufacturer Looks at Pulse Dampeners

A.B. Carpenter
Gas Dynamics Engrg., Ingersoll-Rand Co., Painted Post, NY, ASME Paper No. 76-Pet-14

Key Words: Compressors, Pulse excitation, Dampers

This paper presents a discussion on: development of pulse dampeners for reciprocating gas compressors; advent of SGA compressor system analog simulator; early difficulties with dampener simulation; laboratory experiments compared with analog results; problems with industry specifications at high pressures; present-day thinking regarding dampeners at high pressures; pros and cons of orifices including examples in the field; and satisfactory results with analog designed pulse dampeners, and the need for further research.

RAIL

(Also see Nos. 867, 868)

77-1010

Rail Traffic Noise

C. Stüber
8035 Gäuting, Hubertusstrasse 13, Germany, Acustica, 36 (3), pp 192-202 (Nov 1976) 6 figs, 37 refs
(In German)

Key Words: Railroad trains, Traffic noise, Noise measurement

A survey is given of the results of noise measurements of rail vehicles of all kinds. They are drawn from the researches made by the author and his co-workers during his time with the German State Railway as well as from the literature on the subject up to 1975. Up to date findings are stated on the origin of the rail traffic noise, the exterior noise, its maximum level, the energy equivalent of the working sound level, the noise of the environment, the inside noise, the excitation of body noise through the rail vehicles and the possibilities of lessening the rail traffic noise are dealt with in detail.

77-1011

Investigation of the Noise Produced During the Passage of Railway Trains

M. Louden
Ingenieur-Geologisches Institut Dipl.-Ing. S. Niedermeyer, D-8821 Westheim, Acustica, 36 (3), pp 228-232 (Nov 1976) 8 figs, 4 refs
(In German)

Key Words: Railroad trains, Noise generation

The noise from passing trains is investigated and some of its characteristic values are determined. The statistical distribution is given of this noise during 24 hours measuring time. The dependence of the peak level, mean energy level, frequency spectrum and acoustical passage time on the distance to the railway track is also investigated.

77-1012

Mechanics of Train Collision

P. Tong

Transportation Systems Center, Cambridge, MA, Rept. No. DOT-TSC-FRA-76-5, FRA/ORD-76/246, 74 pp (Apr 1976)

PB-258 993/5GA

Key Words: *Collision research (railroad), Mathematical models*

A simple and a more detailed mathematical model for the simulation of train collisions is presented. The study presents considerable insight as to the causes and consequences of train motions on impact. Comparison of model predictions with two full scale train-to-train impact tests shows good correlation. Methods for controlling train motion and kinetic energy dissipation for the minimization of train collision induced damage are suggested.

77-1013

Analytical Model for Guideway Surface Roughness

M.B. Krishna and D.A. Hullender

Dept. of Mech. Engrg., The Univ. of Texas at Arlington, TX, J. Dyn. Syst. Meas. and Control, Trans. ASME, 98 (4), pp 425-431 (Dec 1976) 13 figs, 10 refs

Sponsored by the U.S. Dept. of Transportation

Key Words: Surface roughness, Guideways, Mathematical models

An analytical model is presented for calculating the power spectral density of relatively short wavelength guideway irregularities associated with surface roughness. The results are presented in terms of design tolerances which can be interpreted in terms of the familiar California profile index or in terms of measurable deviations from a straight edge. Digital computer numerical simulation techniques are used to verify the model.

RECIPROCATING MACHINE

77-1014

Frequency Characteristics of Cylindrical Chokes

S. Miyakawa

Ebara Mfg., Co., Ltd., Central Res. Inst. 4720, Fujisawa, Fujisawa-shi, Kanagawa-ken, Japan, Bull. JSME, 19 (137), pp 1302-1309 (Nov 1976) 16 figs, 7 refs

Key Words: Chokes (fuel systems), Frequency response

An experiment was carried out on cylindrical chokes with unnegligible inlet length for the purpose of clarifying their dynamic characteristics. The results were compared with the analytical solutions derived with use of the equivalent viscosity. In the experiment the frequency response was obtained on various chokes with spring loads at their outlets, where the fluctuating pressure at the choke inlet was taken as the input, and that at the outlet and the piston displacement as the output. Further the analysis was extended to the case of two chokes connected in series.

77-1015

Effect of Pistons in Diesel Engine Noise Generation. Part 2

M. Röhrle

Ostfildern, Germany, MTZ Motor-tech. Z., 37 (10), pp 409-412 (Oct 1976) 16 figs

(In German)

Key Words: Diesel engines, Pistons, Noise reduction

In the first part of this paper (MTZ 37 (7-8), pp 277-282, 1976) the generation of piston noise is analyzed and several piston types manufactured by Mahle Co. are described. In part 2, measures for noise reduction are described and their noise reducing effect is evaluated.

ROAD

(Also see Nos. 860, 861, 878, 879)

77-1016

Sled Tests of Three-Point Systems Including Air Belt Restraints

M.J. Walsh

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5852-V-1, DOT-HS-801-939, 164 pp (Aug 1976)

PB-258 312/8GA

Key Words: Collision research (automotive), Anthropomorphic dummies, Human response, Seat belts, Safety restraint systems

The experimental test program included six sled tests using five unembalmed cadavers and one anthropomorphic test dummy to evaluate a standard three point belt system and an air belt which inflates during impact. The tests were conducted on the Calspan HYGE sled facility simulating 30 mph frontal collisions for the three-point belts and 47 mph frontal collisions for the air belt systems.

77-1017

Modular Finite Element Approach to Structural Crashworthiness Prediction

P. Tong and J.N. Rossettos

Transportation Systems Ctr., Kendall Square, Cambridge, MA 02142, Intl. J. Computers and Structures, 7 (1), pp 109-116 (Feb 1977) 6 figs, 8 refs

Key Words: Collision research (automotive), Crashworthiness, Mathematical models, Modular approach, Finite element techniques, Computer programs

A modular approach is formulated for purposes of analyzing vehicle crashworthiness, and to provide the flexibility to model different parts of a vehicle with different levels of sophistication, depending on the type of information one is seeking. The idea is to approximate a vehicle structure by a number of modules, each suitable for modeling particular portions of the structure. Basic elements are used for this purpose and include a beam element which allows large three dimensional rotations and plastic hinge action, a special spring element which can only take either tension or compression, a finite size rigid body element and a general elastic element using modal approximations. Development of such elements is described together with an example of the simulation of the impact of a rail car locomotive into the rear of a caboose.

77-1018

Classification of Automobile Frontal Stiffness/Crashworthiness by Impact Testing

M.O. Ryder

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5714-V-1, DOT-HS-801 966, 549 pp (Aug 1976)

PB-258 302/9GA

Key Words: Collision research (automotive), Crashworthiness

The objective of the program was to identify, based on frontal crash performance, potentially soft, nominal or stiff late model (1973-1975) domestic full-size and intermediate automobiles, and also to determine the most crashworthy vehicles in smaller automobile weight classes.

77-1019

Consumer Information Crash Test Program Prediction of Dynamic Crash Responses for Vehicle and Occupants. Volume 1. Summary Report

N.E. Shoemaker, M.O. Ryder, and N.J. DeLeys

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZT-5561-V-27, DOT-HS-802 010, 47 pp (Sept 1976) (see vol. 2, PB-285 131)

PB-258 130/4GA

Key Words: Collision research (automotive), Crashworthiness

The objectives of the program were to generate experimental test data on recent intermediate size automobiles in the areas of damage susceptibility, crashworthiness and repairability and to demonstrate the capability of existing simulation models for predicting the dynamic responses of the vehicles and occupants.

77-1020

Consumer Information Crash Test Program Prediction of Dynamic Crash Responses for Vehicle and Occupants. Volume II. Technical Report

N.E. Shoemaker, M.O. Ryder, and N.J. DeLeys

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZT-5561-V-26, DOT-HS-802 011, 242 pp (Sept 1976) (see vol. 1, PB-258 130)

PB-258 131/2GA

Key Words: Collision research (automotive), Crashworthiness

This report contains vehicle response prediction models and a simulation of vehicle occupant responses.

77-1021

Crashworthiness of the Subcompact Vehicle

R.B. Tanner

Minicars, Inc., Goleta, CA., Rept. No. DOT-HS-801 969, 298 pp (Aug 1976)

PB-258 293/0GA

Key Words: Collision research (automotive), Crashworthiness

This report examined the crashworthiness of subcompact vehicles both analytically and experimentally. The analytical studies include statistical accident analysis and dynamic response modeling. Experimental testing to determine baseline performance consisted of ten dynamic impacts at various antles and velocities.

77-1022

Predicting the Limiting Performance of Automobile Structural Components under Crash Conditions

S. Chander and W.D. Pilkey

Professional Services Div., Control Data Corp., Rockville, MD, Rept. No. DOT-HS-802 007, 131 pp (Sept 1976)

PB-257 443/2GA

Key Words: Collision research (automotive), Testing techniques, Computer programs

The document is the final report of an investigation for determining the optimum performance characteristics of automobiles under collision circumstances. A new technique called limiting performance was used for this purpose.

77-1023

Crash Tests of Five-Foot Radius Plate Beam Guardrail

J.W. Button, E. Buth and R.M. Olson

Texas A&M Research Foundation, College Station, TX, Rept. No. FHWA/RD-76-S0523, 51 pp (June 1975)

Sponsored by the Minnesota Dept. of Highways

PB-258 132/0GA

Key Words: Collision research (automotive), Guardrails

The report relates results of two full-scale vehicle crash tests of a five-foot-radius steel plate beam ('W' beam) end treatment developed by the Minnesota Department of Highways. The test vehicles, which weighed 2250 lbs. and 4500 lbs., were towed head-on into the end of the median rail system at 60 mph.

77-1024

Development of a Unitized School Bus. Volume II. Technical Report

L. Adams, A. Kadikar, L. Pauls and W. Rup

Advanced Systems Lab., AMF, Inc., Goleta, CA., Rept. No. DOT-HS-802 005, 558 pp (Aug 1976) (see also vol. 1, PB-257 654 and vol. 3, PB-258 421)

PB-258 301/1GA

Key Words: Buses, Collision research (automotive)

The development of new design concepts for school bus body structures and for passively restraining school bus passengers were the major objectives of this program. Passenger protection in front and rear rigid barrier impacts and in a side impact with a rigid pole - all at a 30 mph impact velocity - were the design goals.

77-1025

Development of a Front Passenger Aspirator Air Bag System for Small Cars

D.J. Romeo

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5777-V-1, DOT-HS-802 039, 136 pp (Sept 1976)

PB-259 008/1GA

Key Words: Air bags (safety restraint systems), Automobiles

The objective of the study was to adapt the aspirated air bag system to the crashworthy small car front seat passenger.

77-1026

Test Programs to Determine Performance Capability of Wheel/Rim Assemblies

J.E. Shearer

Compliance Testing, Inc., Ravena, OH, Rept. No. DOT-HS-802 032, 49 pp (Sept 1976)

PB-259 009/9GA

Key Words: Testing techniques, Vehicle wheels, Fatigue tests, Cornering effects

The purpose of the investigation was to obtain preliminary data on wheel and rim assemblies for a proposed Federal Motor Vehicle Safety Standard and also to provide verification and validation of test methods used by industry. Three test methods were used: dynamic fatigue test, dynamic cornering fatigue 90 degrees, dynamic cornering fatigue 40 degrees.

ROTORS

(See Nos. 893, 924, 925, 987, 988, 1008)

SHIP

77-1027

Tanker Structural Analysis for Minor Collisions

Rosenblatt (M) and Son, Inc., NY, Rept. No. USCG-D-72-76, 642 pp (Dec 1975)

AD-A031 031/8GA

Key Words: Collision research (ships), Tanker ships

This report describes the work accomplished during the course of the project on the evaluation of tanker structure in collision. The intent of the report is to present the investigations performed in evaluating the phenomena that contribute to the ability of a longitudinally framed ship, particularly a tanker, to withstand a minor collision. A minor collision is one in which the cargo tanks remain intact. The ability to withstand a minor collision is quantized by the total energy that can be absorbed during the collision.

77-1028

Ship Motion Effects in the Human Factors Design of Ships and Shipboard Equipment

R.A. Newman

Navy Personnel Res. and Dev. Ctr., San Diego, CA., Rept. No. NPRDC-TR-77-2, 46 pp (Nov 1976) AD-A031 978/0GA

Key Words: *Ship vibration, Human factors engineering*

Ship motion can seriously degrade task performance even when personnel are not actively sea sick. This report deals with motion effects which can be considered in the design of ships and shipboard equipment. Design guidelines are provided for use in the human factors design of ships and shipboard equipment to help minimize the degradation of task performance due to ship motion. Background material on the mechanisms by which motion affects personnel is also provided as an aid to the human factors engineer in performing his design function.

SPACECRAFT

77-1029

Analysis of Structural Dynamic Data from Skylab. Volume 2: Skylab Analytical and Test Modal Data. Final Report

L. Demchak and H. Harcrow

Martin Marietta Aerospace, Denver, CO, Rept. No. NASA-CR-2728, 213 pp (Aug 1976)
N76-33251

Key Words: Spacecraft, Modal tests

A compendium is presented of orbital configuration test modal data, analytical test modal data, analytical test correlation modal data and analytical flight configuration 1.2 modal data. Section A presents tables showing the generalized mass contributions for each of the thirty test modes. Section B presents the two dimensional mode shape plots for the thirty test modes. Tables of GMC's for the test correlated analytical modes are presented in Section C.

77-1030

Dynamics and Control of Non-Rigid Spacecraft

European Space Agency, Paris, France, Rept. No. ESA-SP-117, 376 pp (July 1976)
N77-10142

Key Words: Spacecraft, Dynamic response, Mathematical models

Topics are presented in the field of flexible spacecraft configurations. Dynamic models are reviewed for spinning and non-spinning satellites. The attitude stability of flexible satellites is discussed. Applications of modern control theory are described. The design of control systems for these configurations is outlined. Test and flight verifications are reported, and some special dynamic problems are covered.

STRUCTURAL

77-1031

Modeling and Dynamic Analysis of Picking Mechanisms of Fly Shuttle Looms

B.M. Patel

Ph.D. Thesis, North Carolina State Univ. at Raleigh, 159 pp, 1976
UM 76-28,508

Key Words: Textile looms, Dynamic response, Mathematical models

Mechanical separation between cam and pick ball in fly shuttle looms gives rise to impact and vibration which causes significant noise emission. A basic analytical investigation is made of this phenomenon using a mathematical model which simulates the dynamic response characteristics of the system. Combination of system parameters and operating conditions which give rise to separation are predicted by examining the contact force developed between cam and pick ball.

77-1032

Seismic Analysis for Alaskan North Slope Construction

C.M. Johnson

Brown & Root, Inc., Bellington, WA, ASME Paper No. 76-Pet-75

Key Words: Pile structures, Seismic design

The type of construction used on the North Slope of Alaska is the modular concept supported by steel pilings. The constructed systems experience high stresses from the small earthquakes to be expected in the region. This paper presents the analysis of the modules for earthquake and other loadings. The earthquake criteria are by N.M. Newmark and follow the conservative criteria used for nuclear power plants. Resulting interaction ratios for the critical sections are high but remain within acceptable limits.

77-1033

Some Results on Dynamic Characteristics of Major Components of 1200-MW Fossil Fuel Steam Generating Plant

T.Y. Yang, C.T. Sun, H. Lo, K.W. Kayser, and J.L. Bogdanoff

Purdue Univ., West Lafayette, IN, ASME Paper No. 76-JPGC-Pwr-11

Key Words: Fossil power plants, Boilers, Piping systems, Coal handling equipment, Cooling towers, Chimneys, Seismic response, Natural frequencies, Mode shapes

Analytical and experimental studies of the dynamical characteristics of a 1200-MW fossil fuel steam generating plant currently in progress are described. Five major components - steam generator, steam piping, coal handling equipment, cooling tower, and chimney - are under study. Analytical results are based on detailed finite element analyses; the experimental studies are conducted at the plant site. Natural frequencies and corresponding mode shapes are presented for steam generator, cooling tower, and chimney. Response of cooling tower and chimney to a seismic disturbance are discussed. Experimental results for the chimney are presented.

77-1034

Parametric and Nonlinear Mode Interaction Behaviour in the Dynamics of Structures

A.D.S. Barr and R.P. Ashworth

Dept. of Mech. Engrg., Dundee Univ., Scotland, Rept. No. AFOSR-TR-76-1131, 26 pp (1976)

AD-A031 323/9GA

Key Words: Internal resonance, Structural response

This report describes the behavior of structures under internal resonance conditions. Completion of the analysis on the three mode interaction case yields the variational equations describing the slowly varying amplitudes and phases of each mode. Computer simulation (C.S.M.P.) of these equations has been carried out. Both parametric and nonlinear terms are found to be of importance in the response of the structure. Further experimental work on the original four mode model, where parametric excitation of two of the modes leads to eventual participation of all the modes, is presented along with a theoretical approach of a more general nature. Attention is then turned to a less complex two-mode system exhibiting both parametric and nonlinear behavior.

TRANSMISSIONS

(See No. 899)

TURBOMACHINERY

(Also see Nos. 894, 895)

77-1035

Computer Method for Design of Acoustic Liners for Turbofan Engines

G.L. Minner and E.J. Rice

NASA, Lewis Research Ctr., Cleveland, OH, Rept. No. NASA-TM-X-3317; E-8428, 93 pp (Oct 1976)

N76-33954

Key Words: Turbofan engines, Acoustic linings, Computer aided techniques, Automated design

A design package is presented for the specification of acoustic liners for turbofans. An estimate of the noise generation was made based on modifications of existing noise correlations, for which the inputs are basic fan aerodynamic design variables.

77-1036

Dynamic Response of Cavitating Turbomachines

S.-L. Ng

Div. of Engrg. and Applied Science, California Inst. of Tech., Pasadena, CA., Rept. No. NASA-CR-150036; E-183.1, 203 pp (Aug 1976)

N77-10540

Key Words: Turbomachinery, Dynamic response, Cavity effect

Stimulated by the pogo instability encountered in many liquid propellant rockets, the dynamic behavior of cavitating inducers is discussed. An experimental facility where the upstream and downstream flows of a cavitating inducer could be perturbed was constructed and tested. The upstream and downstream pressure and mass flow fluctuations were measured. Matrices representing the transfer functions across the inducer pump were calculated from these measurements and from the hydraulic system characteristics for two impellers in various states of cavitation. The transfer matrices when plotted against the perturbing frequency showed significant departure from steady state or quasi-steady predictions especially at higher frequencies.

AUTHOR INDEX

TECHNICAL NOTES

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Effects of Bow Force and Speed on Violin Response
J. Acoust. Soc. Amer., 60 (1), pp 274-275 (July 1976) 2 figs, 3 refs

G.V. Rao, I.S. Raju and K.K. Raju
Nonlinear Vibrations of Beams Considering Shear Deformation and Rotary Inertia
AIAA J., 14 (5), pp 685-687 (May 1976) 5 refs

S. El-Asfouri, O. Johnson and B. Mcinnis
On the Problem of Minimizing the Volume of a Vibrating Cantilever
Quart. Appl. Math., 34 (2), pp 203-206 (July 1976) 1 fig, 5 refs

M. Nikkhah-Bahrami
Formulation of the Eigenvalue Problem for an Unconstrained Circular Bar
J. Appl. Mech., Trans. ASME, 43 (1), pp 166-168 (Mar 1976) 2 figs, 9 refs

B. Bhat Rama and H. Wagner
Natural Frequencies of a Uniform Cantilever with a Tip Mass Slender in the Axial Direction
J. Sound Vib., 45 (2), pp 304-307 (Mar 1976) 1 fig, 4 refs

P.A.A. Laura and R. Gelos
Fundamental Frequency of Vibration of a Circular Plate Elastically Restrained Against Rotation and Carrying a Concentrated Mass
J. Sound Vib., 45 (2), pp 298-301 (Mar 1976) 2 figs, 2 refs

J. Ramachandran
Large Amplitude Natural Frequencies of Rectangular Plates with Mixed Boundary Conditions
J. Sound Vib., 45 (2), pp 295-297 (Mar 1976) 3 figs, 9 refs

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J. Appl. Mech., Trans. ASME, 43 (1), pp 180-181 (Mar 1976) 1 fig, 3 refs

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Dynamically Coupled Motion of Surface-Fluid-Shell System
J. Appl. Mech., Trans. ASME, 43 (3), pp 507-508 (Sept 1976) 5 refs

K.K. Raju and G.V. Rao
Non-Linear Vibrations of Orthotropic Plates by a Finite Element Method
J. Sound Vib., 48 (2), pp 301-303 (Sept 1976) 4 refs

W. Gad
New Design of Machine Foundations by Means of Vibration Isolation
Bauingenieur, 51 (4), pp 143-145 (Apr 1976) 4 figs

R.A. Ormiston
Application of Simplified Inflow Models to Rotorcraft Dynamic Analysis
J. Amer. Helicopter Soc., 21 (3), pp 34-37 (July 1976) 3 figs, 8 refs

H.C. Huang
Minimum-Weight Design of Vibrating Elastic Structures with Dynamic Deflection Constraint
J. Appl. Mech., Trans. ASME, 43 (1), pp 178-180 (Mar 1976) 9 refs

R.G. Jacquot
On Vibration of Systems with Mixed Boundary Conditions
J. Sound Vib., 48 (3), pp 421-424 (Oct 8, 1976) 13 refs

R.R. Craig, Jr. and C. Chang
Free-Interface Methods of Substructure Coupling for Dynamic Analysis
AIAA J., 14 (11), pp 1633-1635 (Nov 1976) 1 fig, 6 refs

MEETING	DATE	LOCATION	CONTACT
	1977 MAY		
23rd International Instrumentation Symposium	1-5	Las Vegas, NV	ISA Hq.
Symp. on Fatigue Testing of Weldments	1-6	Toronto, Canada	ASTM Hq., Ms. J.B. Wheeler
Offshore Technology Conference	2-5	Houston, TX	Society of Petroleum Engineers 6200 N. Central Expressway Dallas, TX 75206
Symp. on Statistical Design of Fatigue Experiments	5	Toronto, Canada	ASTM Hq., Ms. J.B. Wheeler
American Helicopter Society Annual National Forum	9-11	Washington, D.C.	American Helicopter Society, Exec. Director, 30 E. 42nd St., New York, New York 10017
Design Engineering Conference and Show	9-12	Chicago, IL	ASME Hq.
National Plant Engineering and Maintenance Show and Conference	9-12	Chicago, IL	Clapp & Poliak Banner & Greif Ltd. 369 Lexington Ave. New York, NY 10017
31st Annual Technical Conference, ASQC	16-18	Philadelphia, PA	R.W. Shearman, ASQC Hq.
Society for Experimental Stress Analysis 1977 Spring Meeting & Exposition	15-20	Dallas, TX	SESA Hq., B. E. Rossi
National Aerospace Electronics Conference	17-19	Dayton, OH	NAECON 140 E. Monument Ave. Dayton, OH 45402
Society of Naval Architects and Marine Engineers 1977 Spring Meeting and STAR Symposium	25-27	San Francisco, CA	A. J. Haskell, Matson Navigation Co. 100 Mission St. San Francisco, CA 94105
6th Canadian Congress of Applied Mechanics	30 May - 3 Jun	Vancouver, Canada	Prof. J.P. Duncan, ME Dept. Univ. of British Columbia Vancouver, BC, Canada
Symposium on Tire Vibration and Noise	May		H. G. Schwartz ASTM Subcommittee F-9.93 on Papers & Symposium E. I. duPont 40 Buchtel Ave. Akron, OH 44308
	JUNE		
Fuels and Lubricants Meeting, SAE	7-9	Tulsa, OK	SAE Hq.
Acoustical Society of America, Spring Meeting	7-10	State College, PA	J. C. Johnson, Appl. Res. Lab. Pennsylvania State University Box 30 State College, PA 16801
National Computer Conference	13-16	Dallas, TX	Ms. P. Isaacson University of Texas Box 688 Richardson, TX 75080

MEETING	DATE	LOCATION	CONTACT
	1977 JUNE		
Applied Mechanics Conference ASME	14-16	New Haven, CT	ASME Hq.
Fluids Engineering Conference	15-17	New Haven, CT	ASME Hq.
4th International Conference on Fracture	19-24	Waterloo, Canada	Prof. T. Kawasaki, Sec. Gen. Int'l. Congress of Fracture c/o Dept. of ME Tohoku University Sendai, Japan
Design Automation Conference	20-22	New Orleans, LA	H. Hayman Box 639 Silver Spring, MD 20901
Symposium on Dynamic Tests on Soil & Rock, ASTM	26 Jun - 1 Jul	Denver, CO	ASTM Hq., Ms. J.B. Wheeler
	JULY		
Application of New Signature Analysis Technology Conference	24-29	Rendge, NH	Dr. Sanford S. Cole Engineering Foundation Conferences 345 E. 47th St. New York, NY 10017 Tele. (212) 644-7835
	AUG		
Society of Automotive Engineers	8-11	Vancouver, Canada	SAE Hq., A.L. Weldy
	SEPT		
Energy Technology Conference and Exhibit	18-23	Houston, TX	ASME Hq.
Vibrations Conference, ASME	26-28	Chicago, IL	ASME Hq.
	OCT		
NOISE-CON 77	10-12	Hampton, VA	Conference Secretariat Noise Control Foundation P.O. Box 3469, Arlington Branch Poughkeepsie, NY 12603 Tele. (914) 462-6719
48th Shock and Vibration Symposium	18-20	Huntsville, AL	Henry C. Pusey, Director The Shock and Vibration Information Center, Code 8404 Naval Research Laboratory Washington, D.C. 20375 Tele. (202) 767-3306
	NOV		
Winter Annual Meeting, ASME	27 Nov-2 Dec	Atlanta, GA	ASME Hq.
	DEC		
Sixth Turbomachinery Symposium	6-8	Houston, TX	Dr. M.P. Boyce, Gas Turbine Labs., Mech. Engrg. Dept., Texas A & M College Station, TX 77843

CALENDAR ACRONYM DEFINITIONS AND ADDRESSES OF SOCIETY HEADQUARTERS

AFIPS: American Federation of Information Processing Societies
210 Summit Ave., Montvale, N.J. 07645

AGMA: American Gear Manufacturers Association
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Washington, D.C.

AIAA: American Institute of Aeronautics and Astronautics, 1290 Sixth Ave.
New York, N.Y. 10019

AIChE: American Institute of Chemical Engineers
345 E. 47th St.
New York, N.Y. 10017

AREA: American Railway Engineering Association
59 E. Van Buren St.
Chicago, Ill. 60605

AHS: American Helicopter Society
30 E. 42nd St.
New York, N.Y. 10017

ARPA: Advanced Research Projects Agency

ASA: Acoustical Society of America
335 E. 45th St.
New York, N.Y. 10017

ASCE: American Society of Civil Engineers
345 E. 45th St.
New York, N.Y. 10017

ASME: American Society of Mechanical Engineers
345 E. 47th St.
New York, N.Y. 10017

ASNT: American Society for Nondestructive Testing
914 Chicago Ave.
Evanston, Ill. 60202

ASQC: American Society for Quality Control
161 W. Wisconsin Ave.
Milwaukee, Wis. 53203

ASTM: American Society for Testing and Materials
1916 Race St.
Philadelphia, Pa. 19103

CCCCAM: Chairman, c/o Dept. ME, Univ. Toronto, Toronto 5, Ontario, Canada

IEEE: Institute of Electrical and Electronics Engineers
345 E. 47th St.
New York, N.Y. 10017

IES: Institute Environmental Sciences
940 E. Northwest Highway
Mt. Prospect, Ill. 60056

IFToMM: International Federation for Theory of Machines and Mechanisms, US Council for TMM, c/o Univ. Mass., Dept. ME, Amherst, Mass. 01002

INCE: Institute of Noise Control Engineering
P.O. Box 3206, Arlington Branch,
Poughkeepsie, N.Y. 12603

ISA: Instrument Society of America
400 Stanwix St., Pittsburgh, Pa. 15222

ONR: Office of Naval Research
Code 40084, Dept. Navy, Arlington, Va. 22217

SAE: Society of Automotive Engineers
400 Commonwealth Drive
Warrendale, Pa. 15096

SEE: Society of Environmental Engineers
6 Conduit St.
London W1R 9TG, England

SESA: Society for Experimental Stress Analysis
21 Bridge Sq.
Westport, Conn. 06880

SNAME: Society of Naval Architects and Marine Engineers, 74 Trinity Pl.
New York, N.Y. 10006

SVIC: Shock and Vibration Information Center
Naval Research Lab., Code 8404
Washington, D.C. 20375

URSI-USNC: International Union of Radio Science - US National Committee c/o MIT Lincoln Lab., Lexington, Mass. 02173

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